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(54) ELECTRICALLY-CONTROLLED MONOLITHIC HIGH-PRESSURE OIL PUMP FOR MARINE LOW-SPEED ENGINE

(57) The present disclosure provides an electrically-controlled monolithic high-pressure oil pump for a marine low-speed engine, to improve the oil inlet throttling and inlet oil pressure stability of the high-pressure oil pump. It comprises: a pump body, a pump cover, an oil inlet-outlet valve assembly, a plunger couple, a plunger spring, a lower spring seat assembly, a guide piston assembly and an electrically-controlled proportional valve, wherein the oil inlet-outlet valve assembly comprises: an oil inlet valve assembly and an oil outlet valve assembly; the oil inlet valve assembly comprises: an oil inlet valve seat, an oil inlet valve and an oil inlet valve spring; the oil outlet valve assembly comprises: an oil outlet valve seat, an oil outlet valve, an oil outlet valve spring and an oil outlet valve spring seat; a high-pressure oil outlet chamber is formed between the oil outlet valve seat and the oil inlet valve seat; a high-pressure oil chamber is formed in the plunger couple, and the high-pressure oil chamber communicates with the high-pressure oil outlet chamber through a first oil hole of the oil inlet valve on the oil inlet valve seat; the electrically-controlled proportional valve communicates with an oil inlet hole of the oil inlet valve seat through the first oil hole on the pump body, and the oil inlet hole of the oil inlet valve seat communicates with or is disconnected from the high-pressure oil chamber; the electrically-controlled proportional valve is provided thereon with a cooling circulation oil passage communicating with a cooling oil passage of the pump body, and the cooling circulation oil passage can cool armature and coil of the proportional valve.

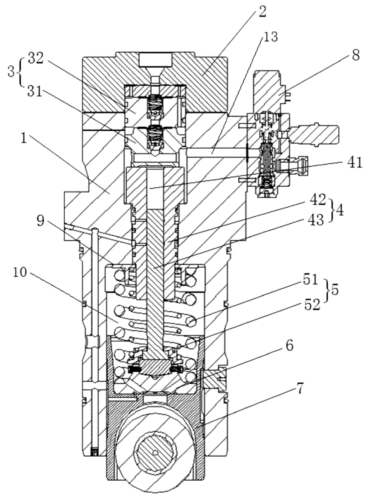

FIG.1

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Description

Technical Field

[0001] The present disclosure relates to the field of high-pressure oil pumps for marine low-speed engines, and in particular, to an electrically-controlled monolithic high-pressure oil pump for a marine low-speed engine.

Background Art

[0002] With the increasingly strict domestic and overseas emission regulations, higher requirements for a fuel injection system are proposed for a marine low-speed diesel engine, and an electrically-controlled common-rail fuel system, which can achieve precise control over fuel injection timing and circulating fuel injection amount, is one of the effective means for high-power marine diesel engines to realize high fuel economy and low emissions of harmful substances. As a common form of electrically-controlled common-rail system oil pumps, electrically-controlled monolithic high-pressure oil pumps are widely used. The conventional proportional valve is generally in a non-cooled structure and cannot satisfy the high-temperature high-viscosity heavy oil usage environment of 750 Cst, therefore, the existing high-pressure oil pumps for low-speed engines are not installed with the proportional valve to improve their working efficiency; a sealing surface of the conventional monolithic high-pressure oil pumps for low-speed engines are mostly special-shaped, which is unfavorable for high-temperature high-pressure heavy oil sealing, has poor reliability, and has relatively high processing difficulty. In the prior art, heavy oil leaked by the plunger couple in the heavy-oil high-pressure oil pumps is mostly discharged into an upper spring cavity of a guide piston and then collected, which design increases the total height of the spring cavity, and at the same time, makes the spring exposed to heavy oil and vulnerable to corrosion.

Summary

[0003] The present disclosure aims at providing an electrically-controlled monolithic high-pressure oil pump for a marine low-speed engine, so as to achieve the effects of improving the oil inlet throttle and the oil inlet pressure stability of the high-pressure oil pump.

[0004] The present disclosure provides an electrically-controlled monolithic high-pressure oil pump for a marine low-speed engine, including:

a pump body, wherein the pump body is provided with a center hole along an axial direction;

a pump cover, wherein the pump cover is mounted on an upper end surface of the pump body;

an oil inlet-outlet valve assembly, a plunger couple, a plunger spring, a lower spring seat assembly and a guide piston assembly, all of which are assembled in the center hole of the pump body; and

an electrically-controlled proportional valve, which is assembled on a side surface of the pump body,

wherein the oil inlet-outlet valve assembly includes: an oil inlet valve assembly and an oil outlet valve assembly;

the oil inlet valve assembly includes: an oil inlet valve seat, an oil inlet valve and an oil inlet valve spring;

the oil inlet valve is mounted in a center hole of the oil inlet valve seat, the oil inlet valve spring is restrained between the oil inlet valve and a bore wall of the oil inlet valve seat, and the oil inlet valve is configured to form a conical seal with the oil inlet valve seat under compression of the oil inlet valve spring;

the oil outlet valve assembly includes: an oil outlet valve seat, an oil outlet valve, an oil outlet valve spring and an oil outlet valve spring seat;

the oil outlet valve spring seat is mounted on an upper end of the oil outlet valve seat, the oil outlet valve is mounted in a center hole of the oil outlet valve seat, the oil outlet valve spring is restrained between the oil outlet valve and the oil outlet valve spring seat, and the oil outlet valve is configured to form a conical seal with the oil outlet valve seat under compression of the oil outlet valve spring;

a high-pressure oil outlet chamber is formed between the oil outlet valve seat and the oil inlet valve seat;

a high-pressure oil chamber is formed in the plunger couple, and the high-pressure oil chamber communicates with the high-pressure oil outlet chamber through a first oil hole of the oil inlet valve on the oil inlet valve seat;

the electrically-controlled proportional valve communicates with an oil inlet hole of the oil inlet valve seat through the first oil hole on the pump body, and the oil inlet hole of the oil inlet valve seat is configured to communicate with or be disconnected from the high-pressure oil chamber; and

the electrically-controlled proportional valve is provided thereon with a cooling circulation oil passage, wherein cooling oil from a cooling oil passage of the pump body, after being injected into the cooling circulation oil passage, is returned to the cooling oil passage of the pump body.

[0005] Optionally, the plunger couple includes:

a plunger sleeve, which is disposed at a lower end of the oil inlet valve seat; and

a plunger, which is slidably inserted into a center hole of the plunger sleeve, wherein the high-pressure oil chamber is defined by the plunger sleeve, the plunger and the oil inlet valve seat,

wherein an inner wall of the plunger sleeve is provided with a first annular groove and a second annular groove;

the pump body is provided with a mixed oil outlet passage and a lubricating oil supply passage, wherein the mixed oil outlet passage communicates with the first annular groove through a mixed oil passage on the plunger sleeve, and the lubricating oil supply passage communicates with the second annular groove through the lubricating oil passage on the plunger sleeve; and

the first annular groove is located above the second annular groove.

[0006] Optionally, the lower spring seat assembly is disposed below the plunger couple, and the lower spring seat assembly includes:

an outer spring seat, wherein the outer spring seat is as a whole of a boss type structure having a central part thick and an outer side thin, and an upper end surface of the outer spring seat is provided with a counterbore having a concave spherical surface;

an upper sphere, wherein a lower portion of the upper sphere is mounted in the counterbore, and a lower end surface of the upper sphere is provided with a convex spherical surface matched with the concave spherical surface; and

an inner spring seat, wherein the inner spring seat is sheathed on an upper portion of the upper sphere, and the inner spring seat has an axial through hole penetrating upper and lower end surfaces thereof,

wherein a lower cylindrical head of the plunger is restrained in the axial through hole, and a lower end surface of the lower cylindrical head of the plunger abuts against an upper end surface of the upper sphere.

[0007] Optionally, a spherical hole is provided at a center of the counterbore, a third annular groove is provided on a lower end surface of the outer spring seat, and the spherical hole and the third annular groove communicate with each other through a lubricating oil inlet passage;
an outer surface of the outer spring seat is a conical surface, the conical surface is provided with a lubricating oil outlet passage, and the lubricating oil outlet passage communicates with a lower end surface of the outer spring seat, and the lubricating oil outlet passage is arranged obliquely;
the upper sphere is provided with a circumferential annular groove in a circumferential direction thereof;
a positioning screw is mounted in the circumferential annular groove after passing through a positioning screw hole of the outer spring seat; and
a distance between an upper surface and a lower surface of the circumferential annular groove is greater than a cylindrical diameter of a portion of the positioning screw located in the circumferential annular groove.

[0008] Optionally, eight lubricating oil outlet passages are specifically provided, and the eight lubricating oil outlet passages respectively communicate with a bottom end surface of the outer spring seat.

[0009] Optionally, the axial through hole provided inside the inner spring seat includes:

a first hole, a second hole and a third hole that have diameters respectively, wherein one diameter is less than another in sequence from top to bottom,

wherein a first guide hole having a gradually increasing diameter is provided between the second hole and the third hole;

a side of the third hole facing the upper sphere is provided with a second guide hole having a gradually increasing diameter;

hole walls of the first guide hole and the second guide hole are formed as guide conical surfaces; and

a part of an upper portion of the upper sphere penetrating the second guide hole is located in the third hole;

the upper sphere and the third hole have a gap of greater than or equal to 1 mm therebetween, and

the counterbore and the upper sphere have a gap of greater than or equal to 1 mm therebetween.

[0010] Optionally, an outer peripheral surface of the inner spring seat and the hole wall of the second hole are each provided with a relief groove; and the upper end surface of the inner spring seat is provided with a weight-reduction annular groove.

[0011] Optionally, the oil pump further includes:

an upper spring seat, which is sheathed on the plunger sleeve and located at an upper end of the inner spring seat;

the plunger spring includes:

a first plunger spring, which is press-fitted between the upper spring seat and the outer spring seat; and

a second plunger spring, which is press-fitted between the upper spring seat and the inner spring seat.

[0012] Optionally, the diameter of the concave spherical surface in the outer spring seat and the diameter of the convex spherical surface of the upper sphere are 20 to 100 times the diameter of the plunger.

[0013] Optionally, the guide piston assembly includes:

a guide piston, wherein the guide piston is provided with a first mounting hole at a central position of an upper end surface thereof, and a second mounting hole on a lower end surface thereof, the first mounting hole and the second mounting hole communicate with each other through a communication hole, and the lower spring seat assembly is mounted in the first mounting hole;

a roller assembly, including a roller mounted in the second mounting hole, a roller bushing interference-assembled in the roller, and thrust sheets interference-assembled at two axial ends of the roller, wherein an annular groove is provided in an axial direction of the roller, and a circular arc transition connection is formed between a groove bottom of the annular groove and an axial end surface of the roller; and

a roller pin, which is clearance-assembled in the roller bushing,

wherein the hole wall of the second mounting hole is provided with a boss, and the boss is in contact with the thrust sheets; and

the boss is uniformly provided with a plurality of first radial oil grooves along a radial direction, and the length directions of first radial oil grooves are in radial directions of the thrust sheets.

[0014] Optionally, an outer surface of the roller pin is provided as a cylindrical surface, a first waist-shaped groove and a second waist-shaped groove are provided at each of two positions on the cylindrical surface, and the first waist-shaped groove and the second waist-shaped groove are provided at a middle of the roller pin; a small-angle wedge groove with an angle between 5 ° and 10 ° is formed between the first waist-shaped groove and an outer surface of the roller bushing, and the second waist-shaped groove is provided therein with a second oil hole; and two second oil holes at the two positions communicate through a lubricating oil outlet passage with each other, and the two second oil holes are arranged with 90 ° therebetween.

[0015] Optionally, an outer surface of the guide piston is provided as a cylindrical surface, on which a circumferential oil groove, a partial circumferential oil groove, a first axial oil groove and a vertical groove are provided, wherein the vertical groove is provided in the circumferential oil groove, and the vertical groove communicates with the partial circumferential oil groove through the first axial oil groove;
the cylindrical surface is further provided with an inclined hole, and two ends of the inclined hole respectively communicate with the inner walls of the circumferential oil groove and the second mounting hole;
the cylindrical surface is further provided with a second axial oil groove communicating with the circumferential oil groove;
the cylindrical surface is further provided with a first straight hole and a second straight hole connected with each other, the first straight hole communicates with the first axial oil groove, and the second straight hole communicates with the first mounting hole; and
the roller pin is provided with a lubricating oil inlet passage on an outer peripheral surface thereof, the lubricating oil inlet passage is provided opposite to the inclined hole, and the lubricating oil inlet passage communicates with the lubricating oil outlet passage.

[0016] Optionally, the lubricating oil inlet passage includes a third radial oil passage disposed along a radial direction of the roller pin and an axial oil passage disposed along an axial direction of the roller pin, wherein the third radial oil passage is connected to the axial oil passage; and the axial oil passage is connected to the oil hole in the second waist-shaped groove.

[0017] Optionally, the roller pin is provided with a DLC coating on the outer peripheral surface thereof;
the roller bushing is made of a copper alloy;
the thrust sheets are each made of a copper alloy;
forced lubrication is adopted between the roller pin and the roller bushing; and
forced lubrication is adopted between the thrust sheets and the boss.

[0018] Optionally, a first chamfer is disposed at each of an upper-end-surface outer periphery and a lower-end-surface outer periphery of the guide piston and the circumferential annular groove;
the hole wall of the first mounting hole is provided with a second chamfer;
the hole wall of the second mounting hole is provided with a fourth hole;
a fifth hole is provided on the outer peripheral surface of the roller pin; and
a return spring and a stop pin are disposed in sequence in the fifth hole, and the stop pin partially extends into the fourth hole.

[0019] Optionally, the lubricating oil inlet passage includes a third radial oil passage disposed along a radial direction of the roller pin and an axial oil passage disposed along an axial direction of the roller pin, wherein the third radial oil passage is connected to the axial oil passage; and the axial oil passage is connected to the second oil hole in the second waist-shaped groove.

[0020] The beneficial effects of the present disclosure are as follows:

(1) The electrically-controlled proportional valve is applied to adjust oil inlet of heavy oil, which can solve the technical problem that the existing mechanical adjustment mode is not high in flexibility. Specifically, the cooling circulation oil passage is provided inside the electrically-controlled proportional valve, so that the cooling oil flowing in the pump body enters the electrically-controlled proportional valve, to cool the electrically-controlled elements in the electrically-controlled proportional valve in a targeted manner, to keep the electrically-controlled elements of the electrically-controlled proportional valve within a normal temperature range, thereby allowing the electrically-controlled proportional valve to perform oil inlet throttling on the pump. The electrically-controlled proportional valve overcomes the disadvantages of mechanical adjustment of oil amount, improves the accuracy, flexibility and response speed of the adjustment of flow of supplied oil, further achieves more accurate matching between the pumped oil supply amount and the operating condition of the diesel engine, avoids degradation of performance caused by insufficient oil supply, also reduces excessive flow during operation, and further reduces the actual load of the pump.

(2) By adding the oil inlet valve assembly, when changing from oil absorption to compression, the high-pressure oil chamber of the plunger sleeve is quickly closed, thus ensuring that the pressure at relevant position in the oil inlet passage of the oil inlet valve seat is stable, and effectively preventing cavitation.

(3) A small amount of lubricating oil in the second annular groove of the plunger couple can be used to completely prevent heavy oil leakage, and prevent the leaked heavy oil from corroding important components such as the plunger spring below the plunger sleeve. In addition, in the present disclosure, sealing the heavy oil with a small amount of lubricating oil can effectively reduce the vertical height of the guide piston (i.e., not needing a relatively long sealing section provided on the guide piston of the low-speed engine in the prior art), further the pump vertical height of the high-pressure oil pump is reduced, and the overall weight of the high-pressure oil pump is reduced. It is tested that the vertical height of the high-pressure oil pump is reduced by 1/3 in the solution of the present disclosure.

(4) The outer spring seat is as a whole of a boss type structure having a central part thick and an outer side thin, the outer spring seat mainly bears the pressure transmitted from the plunger to the upper sphere during operation, and a stress field caused by the pressure is in conical distribution in the outer spring seat. By arranging the outer spring seat to be in a corresponding boss shape, the mass of the outer spring seat can be reduced under the premise of satisfying the strength, thereby reducing the movement mass, and the thicker part of the boss also provides a design space for a middle spherical surface and the oil passage thereof.

(5) The outer spring seat is in spherical fit with the upper sphere, wherein when the lower spring seat assembly with a spherical surface is provided between the plunger and the guide piston, even if there is a relatively large parallelism error between the upper end surface of the guide piston and the tail end surface of the plunger, the spherical surface can be automatically adjusted in angle, so that contact surfaces between the upper sphere and the outer spring seat are kept in sufficient contact, thus eliminating local contact, equalizing the overall force, and relieving the tendency of too large local stress. Meanwhile, a resultant force passes through the center of the spherical center, then the bending moment of accessory is eliminated, further optimizing the dynamic characteristics, and improving the bearing capacity of the system.

(6) The spherical hole provides the lubricating oil to the spherical surface for lubricating the spherical surface, the elastohydrodynamic lubrication effect is formed on the spherical surface using the lubricating oil, thus reducing the wear rate, reducing the contact stress, reducing the fretting damage, and improving the spherical bearing capability and fatigue strength.

(7) An outer surface of the outer spring seat is a conical surface, the lubricating oil outlet passage is provided on the outer conical surface of the outer spring seat, then the plunger spring can be prevented from covering a flow area of the lubricating oil outlet passage, so that the flow area is not affected by the position of the plunger spring.

(8) The inner spring seat is provided with a guide conical surface, which can improve the centering, and the inner spring seat and the upper sphere can be automatically aligned even during striking, thereby improving the tendency of uneven stress.

(9) With the communication hole provided inside the guide piston, when flowing downwards from the com-

munication hole, the lubricating oil above the guide piston is uniformly distributed right at middle of and directly above the second mounting hole of the roller, the lubricating oil is uniformly distributed on the bus bar of the roller, and the distribution of the lubricating oil on the surface of the roller is not affected by the forward and reverse rotation (i.e. the lubricating oil always can be uniformly distributed); the vertical force distribution of the guide piston is improved, i.e. the pressure of the plunger is distributed to a thicker portion around the communication hole, so as to equalize the overall stress, reduce the maximum stress, and improve the reliability of the system bearing capacity; and in the pump assembly, when the guide piston and the outer spring seat cooperate, the lubricating oil outlet passage of the outer spring seat communicates with a place which is above the plunger couple and where the leaked lubricating oil is located, thereby the spring can be prevented from blocking the oil hole, and the flow area of the lubricating oil is increased.

(10) The first radial oil groove provided on the boss is filled up with lubricating oil, and provides sufficient lubricating oil for the moving surface (the end surface of the roller assembly), a dynamic pressure lubricant film is formed on the end surface of the roller using the moving speed of the end surface of the roller, to separate the boss of the guide piston from the end surface of the roller assembly, reduce wear, and reduce the coefficient of friction. The first radial oil groove is provided on the boss of the guide piston. Compared with the case where the first radial oil groove is provided on the roller assembly, the guide piston does not rotate relatively, the high- and low-pressure lubricant film areas on the friction surface are distributed relatively stationary, and the roller assembly thereby is relatively stationary in the axial direction thereof.

(11) The waist-shaped grooves provided in the roller pin have an included angle of 70-120 °, and are located directly above a pressure-bearing region, so as to reduce the influence of providing the waist-shaped groove on the surface on the pressure-bearing region under the condition of sufficiently supplying oil to the friction surface, and further result in a larger angle of the pressure-bearing region and a smaller average pressure of the lubricant film in the pressure-bearing region; the first waist-shaped groove and the corresponding friction surface form a small-angle convergent wedge shape, to enhance the extrusion effect in the dynamic pressure lubrication; the second waist-shaped groove is mainly used to store more lubricating oil, and guarantee sufficient oil supply to the friction surface, then the lubrication on the surface of the roller pin is not affected even if the oil supply is poor within a short period of time, and when the lubrication system is out of work, the probability of the system seizing is reduced.

Brief Description of Drawings

[0021] In order to more clearly illustrate technical solutions of embodiments of the present disclosure, accompanying drawings which need to be used in the embodiments will be introduced briefly below, and it should be understood that the accompanying drawings below merely show some embodiments of the present disclosure, therefore, they should not be considered as limitation on the scope, and those ordinarily skilled in the art still could obtain other relevant accompanying drawings according to these accompanying drawings, without using any creative efforts.

FIG. 1 is a structural schematic view of a high-pressure oil pump provided in embodiments of the present disclosure;

FIG. 2 is a structural schematic view of an oil inlet-outlet valve assembly provided in embodiments of the present disclosure;

FIG. 3 is a structural schematic view of a plunger couple in the prior art;

FIG. 4 is a schematic diagram showing cooperation between the plunger couple and a pump body, and between an oil inlet valve seat and an upper spring seat according to embodiments of the present disclosure;

FIG. 5a is a structural schematic view showing an included angle formed between a plunger and an upper sphere under uneven force according to embodiments of the present disclosure;

FIG. 5b is a structural schematic view showing cooperation between the plunger and the upper sphere after spherical adjustment according to embodiments of the present disclosure;

FIG. 6 is a structural schematic view showing cooperation between a lower spring seat assembly and the plunger according to embodiments of the present disclosure;

FIG. 7 is a schematic sectional view of the lower spring seat assembly provided in embodiments of the present disclosure;

FIG. 8 is a structural sectional view of the lower spring seat assembly provided in embodiments of the present disclosure;

FIG. 9 is a structural schematic view of an inner spring seat provided in embodiments of the present disclosure;

FIG. 10 is a structural schematic view of the upper sphere provided in embodiments of the present disclosure;

FIG. 11 is a schematic sectional view of an outer spring seat provided in embodiments of the present disclosure;

FIG. 12 is a schematic sectional view of the outer spring seat provided in embodiments of the present disclosure;

FIG. 13 is a schematic sectional view of a guide piston assembly provided in embodiments of the present disclosure;

FIG. 14 is a schematic sectional view of a guide piston provided in embodiments of the present disclosure;

FIG. 15 is a structural schematic view of the guide piston provided in embodiments of the present disclosure;

FIG. 16 is a schematic sectional view of the guide piston provided in embodiments of the present disclosure;

FIG. 17 is a structural schematic view of a roller pin provided in embodiments of the present disclosure;

FIG. 18 is a schematic view of axial section of the roller pin provided in embodiments of the present disclosure;

FIG. 19 is a schematic view of radial section of the roller pin provided in embodiments of the present disclosure;

FIG. 20a is a schematic view showing force distribution of a roller assembly, which is not provided with an annular groove, according to embodiments of the present disclosure; and

FIG. 20b is a schematic view showing force distribution of the roller assembly, which is provided with an annular groove, according to embodiments of the present disclosure.

[0022] Illustration of reference signs: 1-pump body; 12-lubricating oil supply passage; 13-first oil hole; 2-pump cover; 3-oil inlet-outlet valve assembly; 31-oil inlet valve assembly; 311-oil inlet valve seat; 312-oil inlet valve; 313-oil inlet valve spring; 32-oil outlet valve assembly; 321-oil outlet valve seat; 322-oil outlet valve; 323-oil outlet valve spring; 324-oil outlet valve spring seat; 33-high-pressure oil outlet chamber; 4-plunger couple; 41-high-pressure oil chamber; 42-plunger sleeve; 421-first annular groove; 422-second annular groove; 423-mixed oil passage; 424-lubricating oil passage; 43-plunger; 431-lower cylindrical head; 5-plunger spring; 51-first plunger spring; 52-second plunger spring; 6-lower spring seat assembly; 61-outer spring seat; 611-counterbore; 612-spherical hole; 613-third annular groove; 614-lubricating oil inlet passage; 615-lubricating oil outlet passage; 616-positioning screw hole; 62-upper sphere; 621-circumferential annular groove; 63-inner spring seat; 631-axial through hole; 6311-first hole; 6312-second hole; 6313-third hole; 6314-first guide hole; 6315-second guide hole; 6316-guide conical surface; 6317-relief groove; 6318-weight-reduction annular groove; 64-positioning screw; 7-guide piston assembly; 71-guide piston; 711-first mounting hole; 7110-second chamfer; 712-second mounting hole; 7121-boss; 7122-first radial oil groove; 7123-fourth hole; 713-communication hole; 714-partial circumferential oil groove, 715-circumferential oil groove; 716-first axial oil groove; 717-vertical groove; 718-inclined hole; 719-second axial oil groove; 7100-first straight hole; 7101-second straight hole; 7102-first chamfer; 72-roller assembly; 721-roller; 7211-annular groove; 722-roller bushing; 723-thrust sheet; 73-roller pin; 731-first waist-shaped groove; 732-second waist-shaped groove; 733-second oil hole; 735-lubricating oil inlet passage; 7351-third radial oil passage; 7352-axial oil passage; 736-fifth hole; 737-return spring; 738-stop pin; 8-electrically-controlled proportional valve; 9-upper spring seat; 10-center hole.

Detailed Description of Embodiments

[0023] Exemplary embodiments of the present disclosure will be described in detail below with reference to accompanying drawings. Although the exemplary embodiments of the present disclosure are shown in the accompanying drawings, it should be understood that the present disclosure can be realized in various forms, but should not be restrained by the embodiments illustrated herein. On the contrary, these embodiments are provided for the purpose of understanding the present disclosure more thoroughly, and being capable of completely conveying the scope of the present disclosure to the person skilled in the art.

[0024] It should be noted that when an element is "fixed" to another element, it may be directly on the another element or there may be an intermediate element therebetween. When one element is "connected" with another element, it may be directly connected to the another element or there may be an intermediate element therebetween. On the contrary, when an element is "directly" "on" another element, there is no intermediate element. Terms used herein such as "perpendicular", "horizontal", "left", "right" and the like are merely for illustrative purpose.

[0025] In the present disclosure, unless otherwise specified and defined explicitly, terms such as "mount", "join", "connect", and "fix" should be construed in a broad sense, for example, a connection may be fixed connection, detachable connection, or integral connection; it may be mechanical connection, or also may be electrical connection; it may be direct connection, indirect connection via an intermediary, or internal communication between two elements or interaction between two elements. For those ordinarily skilled in the art, specific meanings of the above-mentioned terms in the present disclosure can be understood according to specific circumstances.

[0026] Besides, terms "first" and "second" are merely for descriptive purpose, but should not be construed as indicating or implying importance in the relativity or suggesting the number of a related technical feature. Thus, a feature defined with "first" or "second" may explicitly or implicitly mean that one or more such features are included. In the description of the present disclosure, "multiple (a plurality of)" means two or more, unless otherwise defined explicitly.

[0027] Referring to FIG. 1, the present disclosure provides an electrically-controlled monolithic high-pressure oil pump for a marine low-speed engine, including:

a pump body 1, wherein the pump body 1 is provided with a center hole 10 along an axial direction;

a pump cover 2, wherein the pump cover 2 is mounted on an upper end surface of the pump body 1;

an oil inlet-outlet valve assembly 3, a plunger couple 4, a plunger spring 5, a lower spring seat assembly 6 and a guide piston assembly 7, all of which are assembled in the center hole of the pump body 1; and

an electrically-controlled proportional valve 8, which is assembled on a side surface of the pump body 1.

[0028] The oil inlet-outlet valve assembly 3 includes: an oil inlet valve assembly 31 and an oil outlet valve assembly 32.

[0029] The oil inlet valve assembly 31 includes: an oil inlet valve seat 311, an oil inlet valve 312 and an oil inlet valve spring 313.

[0030] As shown in FIG. 2, the oil inlet valve 312 is mounted in the center hole of the oil inlet valve seat 311; the oil inlet valve spring 313 is restrained between the oil inlet valve 312 and a bore wall of the oil inlet valve seat 311; and the oil inlet valve 312 is configured to form a conical seal with the oil inlet valve seat 311 under the compression of the oil inlet valve spring 313.

[0031] The oil outlet valve assembly 32 includes: an oil outlet valve seat 321, an oil outlet valve 322, an oil outlet valve spring 323 and an oil outlet valve spring seat 324.

[0032] The oil outlet valve spring seat 324 is mounted on an upper end of the oil outlet valve seat 321; the oil outlet valve 322 is mounted in the center hole of the oil outlet valve seat 321; the oil outlet valve spring 323 is restrained between the oil outlet valve 322 and the oil outlet valve spring seat 324; and the oil outlet valve 322 is configured to form a conical seal with the oil outlet valve seat 321 under the compression of the oil outlet valve spring 323.

[0033] A high-pressure oil outlet chamber 33 is formed between the oil outlet valve seat 321 and the oil inlet valve seat 311.

[0034] A high-pressure oil chamber 41 is formed in the plunger couple 4, and the high-pressure oil chamber 41 communicates with the high-pressure oil outlet chamber 33 through a first oil hole 13 of the oil inlet valve on the oil inlet valve seat 311.

[0035] The electrically-controlled proportional valve 8 communicates with an oil inlet hole of the oil inlet valve seat 311 through the first oil hole 13 on the pump body 1, and the oil inlet hole of the oil inlet valve seat is configured to communicate with or be disconnected from the high-pressure oil chamber 41. When the plunger moves downwards, the oil inlet valve 312 is opened, matched conical surfaces of the oil inlet valve and the oil inlet valve seat are separated, and the oil inlet hole of the oil inlet valve seat can communicate with the high-pressure oil chamber 41 through a gap between the oil inlet valve seat and the oil inlet valve, an inclined hole of the oil inlet valve, the first oil hole of the oil inlet valve, a gap between the oil inlet valve seat and the oil outlet valve seat, and a vertical hole of the oil inlet valve seat. When the plunger moves upwards, the oil inlet valve is closed, conical surfaces of the oil inlet valve and the oil inlet valve seat are attached to each other, the oil inlet hole of the oil inlet valve seat and the inclined hole of the oil inlet valve are separated by the above conical surfaces, and further the oil inlet hole of the oil inlet valve seat is disconnected with the high-pressure oil chamber 41.

[0036] The electrically-controlled proportional valve 8 is provided thereon with a cooling circulation oil passage configured to make cooling oil from a cooling oil passage of the pump body 1 return, after being injected into the cooling circulation oil passage, to the cooling oil passage of the pump body 1.

[0037] As shown in FIG. 1, the center hole provided in the pump body 1 is a through hole penetrating both upper and lower end surfaces of the pump body 1. The pump cover 2 is fixed to the upper end surface of the pump body 1, and a mounting hole opposite to the center hole of the pump body 1 is provided in a direction of the pump cover 2 facing the pump body 1, and the oil outlet valve seat 321 is mounted in the center hole of the pump body 1 and the mounting hole of the pump body 1.

[0038] As seen from FIG. 1, the oil outlet valve assembly 32 is mounted above the oil inlet valve assembly 31, and above the pump cover 2, there is an oil passage communicating with the oil outlet valve assembly 32, and finally, the high-pressure heavy oil pumped out by the high-pressure oil pump is discharged through the oil passage on the pump cover 2.

[0039] The electrically-controlled proportional valve 8, as a hydraulic control device, has the effect of oil inlet throttling, and the electrically-controlled proportional valve 8 is mainly used for oil inlet regulation of light-weight oil (such as gasoline and light-weight diesel). In the prior art, there is yet no solution to apply the electrically-controlled proportional valve 8 to the oil inlet regulation of heavy oil, for the reason that in operation, the temperature of the heavy oil may be as high as 160 °C, which temperature has exceeded limit operating temperatures of electrically-controlled elements such as armature and coil of the existing electrically-controlled proportional valves 8. In the prior art, for the oil inlet throttling adjustment of the high-pressure oil pumps using heavy oil, a mechanical transmission design is adopted, that is, the oil amount is controlled by means of a spiral groove above a plunger and a speed regulator, while such oil inlet adjusting manner has the disadvantages of low adjusting accuracy of oil amount, slow response to adjustment control, and dependency of oil amount on rotational speed of the speed regulator, etc.

[0040] In an embodiment of the present disclosure, the electrically-controlled proportional valve 8 is applied to adjust oil inlet of heavy oil, which can solve the technical problem that the temperature adjustment flexibility of the existing mechanical adjustment mode is not high. Specifically, a cooling circulation oil passage is provided inside the electrically-controlled proportional valve 8, so that the cooling oil flowing in the pump body 1 enters the electrically-controlled proportional valve 8, to cool the electrically-controlled elements in the electrically-controlled proportional valve 8 in a targeted manner, so that the temperature of electrically-controlled elements of the electrically-controlled proportional valve 8 is kept within a normal range. The cooling circulation oil passage provided in the electrically-controlled proportional valve 8 should satisfy the following requirements: (1) the cooling circulation oil passage should be as close as possible to the electrically-controlled elements of the electrically-controlled proportional valve, such as coil and armature; and (2) the flow rate of the cooling oil introduced into the cooling circulation oil passage should be able to reduce the temperature of the electrically-controlled elements such as coil and armature to be within the operating temperature range. In order to enable the cooling circulation oil passage to meet the requirements, simulation calculation and experiments need to be performed in advance on armatures of different models, to determine specific parameter information such as the spatial arrangement and size of the cooling circulation oil passage in each model. In the above, for the simulation calculation and experiments for armatures of different models, reference may be made to the method in the prior art, and details are not repeated herein.

[0041] The advantages of the above design lie in that the temperature of the armature and the coil of the electrically-controlled proportional valve 8 in the operating state is reduced by the cooling circulation oil passage provided in the electrically-controlled proportional valve 8, so that the electrically-controlled elements operate in the normal temperature range, thereby allowing the electrically-controlled proportional valve 8 to perform oil inlet throttling on the pump. With the use of the electrically-controlled proportional valve 8, the disadvantage of mechanical adjustment of oil amount is overcome, the accuracy, flexibility and response speed of the adjustment of flow of supplied oil are improved, so as to further achieve more accurate matching between the pumped oil supply amount and the operating condition of the diesel engine, avoid degradation of performance caused by insufficient oil supply, also reduce excessive flow during operation, and further reduce the actual load of the pump.

[0042] As shown in FIG. 2, in an oil inlet phase, the inlet valve 312 is configured to be opened under the effect of oil inlet pressure of the electrically-controlled proportional valve 8 and the thrust of the oil inlet valve spring, the oil outlet valve 322 is configured to be sealed with the oil outlet valve seat 321 under the effect of back pressure from the oil flow flowing out therethrough, the low-pressure heavy oil enters the high-pressure oil chamber 41 from the oil inlet hole of the electrically-controlled proportional valve 8, to start an oil inlet operation, and the oil inlet amount is controlled by adjusting the opening degree of the electrically-controlled proportional valve 8, so as to satisfy the requirements of different operating conditions of the high-pressure oil pump, wherein in the oil pumping phase: the guide piston assembly 7 moves upwards, the plunger 43 compresses heavy oil in the high-pressure oil chamber 41, then the pressure of the heavy oil gradually increases, and when the fuel pressure in the high-pressure oil chamber 41 is greater than the oil inlet pressure, the oil inlet valve 312 is closed, and as the high-pressure oil outlet chamber 33 is connected to the high-pressure oil chamber 41, when the fuel pressure in the high-pressure oil chamber 41 exceeds the back pressure and the force of the oil outlet valve spring 323, the oil outlet valve 322 is opened, and the high-pressure fuel is discharged from the center hole of the pump cover 2 through the oil outlet valve spring seat 324. FIG. 3 shows the structure of the plunger couple in the prior art, and as shown in FIG. 3, the high-pressure common-rail heavy oil pump in the prior art adopts a mechanical design, the oil inlet passage 505 is provided on a plunger sleeve 600, the plunger 500 is slidably inserted into the plunger sleeve 600, and no oil inlet valve assembly is provided. In operation, during alternation from oil absorption to compression, a part of the pressurized fuel will flow from the oil inlet passage 505 back to the low-pressure oil inlet passage, which in turn results in a large pressure change in the oil inlet passage 505, thus easily causing cavitation at relevant position on the oil inlet passage 505. This is also one of the main destructive forms of the plunger couple observed in practical marine experiments. Compared with the structure of high-pressure oil pump in the prior art, in the high-pressure oil pump provided in the present disclosure, by adding the oil inlet valve assembly 31, when changing from oil absorption to compression, the high-pressure oil chamber of the plunger sleeve 42 is quickly closed, thus ensuring that the pressure at relevant position in the oil inlet passage of the oil inlet valve seat 311 is stable, and effectively preventing cavitation.

[0043] FIG. 4 shows cooperative relationship between the plunger coupling 4 and the pump body 1, and between the oil inlet valve seat 311 and an upper spring seat 9, and with reference to FIG. 4, the plunger couple 4 includes:

a plunger sleeve 42, which is disposed at a lower end of the oil inlet valve seat 311; and

a plunger 43, which is slidably inserted into the center hole of the plunger sleeve 42, wherein the high-pressure oil chamber 41 is defined by the plunger sleeve 42, the plunger 43 and the oil inlet valve seat 311.

[0044] An inner wall of the plunger sleeve 42 is provided with a first annular groove 421 and a second annular groove 422.

[0045] Optionally, the pump body 1 is provided with a mixed oil outlet passage (not shown in the drawing) and a lubricating oil supply passage 12, wherein the mixed oil outlet passage communicates with the first annular groove 421 through a mixed oil passage 423 on the plunger sleeve 42, the mixed oil formed at the first annular groove 421 flows out to a waste oil tank (not shown in the drawing) through the mixed oil outlet passage and the mixed oil passage 423, and the lubricating oil supply passage 12 communicates with the second annular groove 422 through the lubricating oil passage 424 on the plunger sleeve 42.

[0046] The first annular groove 421 is located above the second annular groove 422. It should be noted that "above" herein is defined based on the positional relationship in the drawing, but does not mean that the horizontal height of the first annular groove 421 must be greater than that of the second annular groove 422 in practical application.

[0047] The effects of the lubricating oil entering the second annular groove 422 include: 1. isolating the fuel entering into a gap between the plunger 43 and the plunger sleeve 42 from the high-pressure oil chamber 41 above the plunger 43, which can prevent the fuel from flowing into transmission components below the plunger 43, and prevent the fuel from invading the transmission components below the plunger 43 to contaminate the lubricating oil system of the whole engine; and 2. allowing all of the friction surfaces below the plunger 43 to be in a lubricating state with clean lubricating oil, and improving the friction state of the plunger 43, wherein compared with the upper heavy oil, the lubricating oil has higher cleanliness, and the lubricating oil contains an additive that improves friction, and can form a better lubricant film compared with lubrication with the heavy oil.

[0048] Since the low-speed engine in the prior art allows the heavy oil to leak below the plunger 43, and then the leaked heavy oil is collected separately, the leaked heavy oil is at risk of corroding the plunger spring 5 and other components below the plunger 43. In the high-pressure oil pump provided in the present disclosure, a small amount of lubricating oil in the second annular groove 422 of the plunger sleeve 42 of the plunger couple 4 can be used to completely prevent heavy oil leakage, and prevent the leaked heavy oil from corroding important components such as the plunger spring 5 below the plunger sleeve 42. In addition, for the detects such as relatively high overall vertical height of the high-pressure oil pump and high manufacturing cost caused by the complex dynamic sealing mechanism provided on the guide piston below the plunger of the low-speed engine in the prior art, the high-pressure oil pump provided in the present disclosure can effectively reduce, by sealing the heavy oil with a small amount of lubricating oil, the vertical height of the guide piston 71 (the conventional heavy oil guide piston is provided with a relatively long heavy oil sealing section), further reduce the pump vertical height of the high-pressure oil pump, and reduce the overall weight of the high-pressure oil pump. It is tested that compared with the high-pressure oil pump of the low-speed engine in the prior art, the vertical height of the high-pressure oil pump provided in the present disclosure is reduced by 1/3.

[0049] Optionally, referring to FIG. 1 and FIG. 6 to FIG. 12, the lower spring seat assembly 6 is disposed below the plunger couple 4, and the lower spring seat assembly 6 includes:

an outer spring seat 61, wherein the outer spring seat is as a whole of a boss type structure having a central part thick and an outer side thin, the outer spring seat 61 mainly bears the pressure transmitted from the plunger 43 to the upper sphere during operation, and a stress field caused by the pressure is in conical distribution in the outer spring seat 61. By arranging the outer spring seat 61 to be in a corresponding boss shape, the mass of the outer spring seat 61 can be reduced under the premise of satisfying the strength, thereby reducing the movement mass of the outer spring seat 61, the central thicker part of the boss also provides a design space for a middle spherical surface and the oil passage, an upper end surface of the outer spring seat 61 is provided with a counterbore 611 having a concave spherical surface, the lower spring seat assembly 6 further includes an upper sphere 62, a lower portion of which is mounted in the counterbore 611, and a lower end surface of the upper sphere 62 is provided with a convex spherical surface matched with the concave spherical surface; and

an inner spring seat 63, which is sheathed on an upper portion of the upper sphere 62, wherein the inner spring seat 63 has an axial through hole 631 penetrating upper and lower end surfaces thereof,

wherein a lower cylindrical head 431 of the plunger 43 is restrained in the axial through hole 631, and a lower end surface of the lower cylindrical head 431 of the plunger 43 abuts against the upper end surface of the upper sphere 62.

[0050] It can be seen from experiments that when the plunger 43 is in operation, as there is a parallelism error between a tail plane and a corresponding compression surface (guide piston or spring seat surface), the tail plane of the plunger 43 may be locally stressed too much when being tightly pressed (as shown in FIG. 5a, wherein in FIG. 5a, an included angle of β is formed between the upper sphere 62 and the plunger 43), the uneven force distribution in turn will produce an additional moment applied to the plunger 43, further bringing additional load and energy loss to the system, and affecting the system dynamic characteristics. When a lower spring seat assembly 6 with a spherical surface is provided between the plunger 43 and the guide piston 71, even if there is a relatively large parallelism error between the upper end surface of the guide piston 71 and the tail end surface of the plunger 43, the spherical surface can automatically perform angle adjustment, so that contact surfaces of the upper sphere 62 and the outer spring seat 61 are kept in sufficient contact (the state of FIG. 5a is changed to the state of FIG. 5b, and in FIG. 5b, two contact surfaces of the upper sphere 62 and the plunger 43 are fitted/attached to each other), thus eliminating local contact, equalizing the overall force, and relieving the tendency of too large local stress. Meanwhile, the support force of the upper sphere 63 for the plunger 61 passes through the center of the spherical surface, then the bending moment of accessory is eliminated, further optimizing the dynamic characteristics, and improving the bearing capacity of the system.

[0051] Optionally, referring to FIG. 7 to FIG. 12, the above spherical hole 612 is provided at the center of the counterbore 611, a third annular groove 613 is provided on a lower end surface of the outer spring seat 61, the spherical hole 612 and the third annular groove 613 communicate with each other through a lubricating oil inlet passage 614, the lubricating oil inlet passage 614 communicates with the piston oil passage on the guide piston 71, the lubricating oil forms a lubricant film at the convex spherical surface of the outer spring seat 61 through the lubricating oil inlet passage 614, then fretting corrosion and damage between the convex spherical surface of the upper sphere 62 and the concave spherical surface of the outer spring seat 61 can be effectively prevented; the spherical hole 612 provides the lubricating oil to the spherical surface for lubricating the spherical surface, the elastohydrodynamic lubrication effect is formed on the spherical surface using the lubricating oil, thus reducing the wear rate, reducing the contact stress, reducing the fretting damage, and improving the spherical bearing capability and fatigue strength.

[0052] As shown in FIG. 6 and FIG. 7, an outer surface of the outer spring seat 61 is a conical surface, the conical surface is provided with a lubricating oil outlet passage 615, and the lubricating oil outlet passage 615 communicates with a lower end surface of the outer spring seat 61; the lubricating oil outlet passage 615 is arranged obliquely; the lubricating oil outlet passage may make upper and lower regions of the outer spring seat 61 communicate with each other, so that the lubricating oil above the outer spring seat 61 flows smoothly into the lower part, thereby preventing a lubricating oil chamber above the outer spring seat 61 from being filled up, and compressing the additional load caused by the lubricating oil. The lubricating oil outlet passage 615 is provided on the outer conical surface of the outer spring seat 61, then the plunger spring 5 can be prevented from covering a flow area of the lubricating oil outlet passage 615, so that the flow area is not affected by the position of the plunger spring 5.

[0053] Optionally, as shown in FIG. 8, eight lubricating oil outlet passages 615 may be provided; the eight lubricating oil outlet passages 615 respectively communicates with the bottom end surface of the outer spring seat 61, so as to ensure that the lubricating oil in the upper space of the outer spring seat 61 can flow out smoothly, and the additional load caused by the accumulation of the lubricating oil is avoided; meanwhile, obliquely disposing the lubricating oil outlet passage 615 at the conical surface of the outer spring seat 61 also prevents accumulation caused by unsmooth flow of the lubricating oil due to the fact that the plunger spring 5 blocks the lubricating oil outlet passage 615. It should be noted that a person skilled in the art could adjust the number and diameter of the above lubricating oil outlet passage 615 according to the lubricating oil flow rate and space.

[0054] As shown in FIG. 10, optionally, the upper sphere 62 is provided with a circumferential annular groove 621 in a circumferential direction.

[0055] As shown in FIG. 11, optionally, a positioning screw 64 is mounted in the circumferential annular groove 621 after passing through a positioning screw hole 616 of the outer spring seat 61.

[0056] A distance between an upper surface and a lower surface of the circumferential annular groove 621 is greater than a cylindrical diameter of a portion of the positioning screw 64 located in the circumferential annular groove 621.

[0057] Optionally, the upper sphere 62 and the outer spring seat 61 are connected by a positioning screw 64 with threads, the positioning screw 64 is fixed on the outer spring seat 61 by threads, a head of the positioning screw 64 is set as a cylindrical surface and is a positioning portion, and a corresponding circumferential annular groove 621 is provided on the upper sphere 62 for mounting a pin head. Optionally, the circumferential annular groove 621 on the upper sphere 62 may also be embodied as a circular hole. The positioning screw 64 can approximately position the upper sphere 62 and the outer spring seat 61, and prevent the upper sphere 62 from falling out of the outer spring seat 61 during reciprocation when the plunger 43 and the upper ball 62 are separated from each other.

[0058] Optionally, as shown in FIG. 9, the axial through hole 631 provided inside the inner spring seat 63 includes:

a first hole 6311, a second hole 6312 and a third hole 6313 that have diameters respectively, wherein one diameter is less than another in sequence from top to bottom;

a first guide hole 6314 having a diameter gradually increasing from top to bottom in a vertical direction is provided between the second hole 6312 and the third hole 6313;

one side of the third hole 6313 facing the upper sphere 62 is provided with a second guide hole 6315 having a diameter gradually increasing from top to bottom in the vertical direction;

hole walls of the first guide hole 6314 and the second guide hole 6315 are formed as guide conical surfaces 6316; and

a part of an upper portion of the upper sphere 62 penetrating the second guide hole 6315 is located in the third hole 6313.

[0059] In the above, the upper end surface of the lower cylindrical head 431 of the plunger 43 abuts against the upper end surface of the second hole 6312; and the hole wall of the second hole 6312 is fitted/attached to an annular surface of the lower cylindrical head 431 of the plunger 43. As the hole walls of the first guide hole 6314 and the second guide hole 6315 are formed as guide conical surfaces 6316, if the plunger 43 is separated from the upper sphere 62 or the inner spring seat 63 is separated from the plunger 43, when the plunger 43 strikes against the upper sphere 62 again, the guide conical surface 6316 will automatically align the plunger 43 with the upper sphere 62, align the plunger 43 with the inner spring seat 63, and align the inner spring seat 63 with the upper sphere 62, thus preventing a large angular deviation and radial displacement between the plunger 43 and the inner spring seat 63, and between the inner spring seat 63 and the upper sphere 62, and further the system constituted by the above components also can be ensured to be in an appropriate position even during impact, so that the overall force on the system is equalized. Specifically, when the plunger 43 is seized, the inner spring seat 63 is relatively stationary (i.e. seized at an upper stop point of the plunger 43), and the outer spring seat 61 and the upper sphere 62 will strike against each other in a reciprocating manner. The inner spring seat 63 and the plunger 43 may not be centered with the upper sphere 62 during striking, thus causing the upper sphere 62 to be locally stressed upon striking. The inner spring seat 63 is provided with a guide conical surface 6316, which can improve the centering between the inner spring seat 63 and the plunger 43 and the upper sphere 62, and the inner spring seat 63 and the upper sphere 62 can be automatically aligned even during striking, thereby improving the tendency of uneven stress.

[0060] Optionally, the upper sphere 62 and the third hole 6313 have a gap of greater than or equal to 1 mm therebetween, and optionally, an outer cylindrical surface of the upper sphere 62 and a hole wall of the third hole 6313 have a gap of 1 mm therebetween. As the outer spring seat 61 and the upper sphere 62 are in spherical fit and have a relatively large (millimeter-sized) gap, the outer spring seat and the upper sphere may slide relatively freely, so that in the operation process of the plunger, if there is an angle error between a lower end surface of the lower cylindrical head 431 of the plunger 43 and an upper end surface of the upper sphere 62, the upper sphere 62 will slide, when the plunger 43 moves downwards and strikes against the upper sphere 62, with respect to the outer spring seat 61 to automatically compensate for the angle error, and further, when the plunger 43 is subjected to an additional load, the lower end surface of the lower cylindrical head 431 of the plunger 43 and the upper end surface of the guide piston 71 are uniformly stressed, which can effectively prevent too large local stress.

[0061] Optionally, the counterbore 611 and the upper sphere 62 have a gap of greater than or equal to 1 mm therebetween, and optionally, the outer cylindrical surface of the upper sphere 62 and the cylindrical surface of the counterbore 611 have a gap of 1 mm therebetween, that is, the upper sphere 62 and the inner spring seat 63 have a relatively large (1 mm) gap therebetween, and the positioning screw 64 and the upper sphere 62, the positioning screw 64 and the outer spring seat 61, and the upper sphere 62 and the inner spring seat 63 each have a relatively large (1 mm) gap therebetween, which ensures that the effective rotational freedom of the upper sphere 62 in the radial movement will not be restrained by the positioning screw 64, and the effective rotational freedom of the plunger 43 and the upper sphere 62 in the radial movement will not be restrained by the inner spring seat 63, then the plunger 43 is prevented from being subjected to a radial additional load.

[0062] Optionally, the upper sphere 62 and the third hole 6313, and the counterbore 611 and the upper sphere 62 each have a millimeter-sized gap therebetween, which allows macroscopic angle error of the upper sphere 62 relative to the outer spring seat 61, so that the technical effects of preventing seizing of spherical surface, eliminating local contact, balancing overall stress, and relieving the tendency of too large local stress can be achieved.

[0063] Besides, as shown in FIG. 9, optionally, an outer peripheral surface of the inner spring seat 63 and the hole wall of the second hole 6312 are each formed with a relief groove 6317; and the upper end surface of the inner spring seat 63 is provided with a weight-reduction annular groove 6318 that surrounds a central axis of the spring seat 63.

[0064] Optionally, referring to FIG. 1, the electrically-controlled monolithic high-pressure oil pump for a marine low-speed engine provided in the present disclosure further includes:

an upper spring seat 9, which is sheathed on the plunger sleeve 42, and is located at an upper end of the inner spring seat 63;

the plunger spring 5 includes:

a first plunger spring 51, which is press-fitted between the upper spring seat 9 and the outer spring seat 61; and

a second plunger spring 52, which is press-fitted between the upper spring seat 9 and the inner spring seat 63.

[0065] Optionally, the diameter of the concave spherical surface in the outer spring seat 61 and the diameter of the convex spherical surface of the upper sphere 62 each are 20 to 100 times the diameter of the plunger 43. The parallelism error magnitude of a tail portion of the plunger 43 and the upper end surface of the guide piston 71 is relatively low, generally in the magnitude of 0.01 mm, and the requirement on the spherical angle adjusting capability is low, so that the small-angle spherical adjustment can also meet the angle adjustment requirement. When the concave spherical surface in the outer spring seat 61 and the convex spherical surface of the upper sphere 62 are relatively large, only a small part of the acting force of the two surfaces, when being pressed, is converted into tensile stress, and for metal materials, the compressive strength is generally higher than the tensile strength, and the compressive stress is not easy to cause fatigue, therefore, the proportion of the tensile stress can be reduced by selecting a large spherical surface, and further the bearing capacity and the fatigue strength of the material are improved.

[0066] Optionally, referring to FIG. 13 to FIG. 19, the guide piston assembly 7 includes:

a guide piston 71, wherein the guide piston is provided with a first mounting hole 711 at a central position of an upper end surface thereof, and a second mounting hole 712 on a lower end surface thereof, and the first mounting hole 711 and the second mounting hole 712 communicate with each other through a communication hole 713, and the lower spring seat assembly 6 is mounted in the first mounting hole 711;

a roller assembly 72, including a roller 721 mounted in the second mounting hole 712, a roller bushing 722 interference-assembled in the roller 721, and thrust sheets 723 interference-assembled at two axial ends of the roller 721, wherein an annular groove 7211 is provided in the axial direction of the roller 721, and a circular arc transition connection is formed between a groove bottom of the annular groove 7211 and an axial end surface of the roller 721; and

a roller pin 73, which is clearance-assembled in the roller bushing 722.

[0067] The hole wall of the second mounting hole 712 is provided with a boss 7121, and the boss 7121 is in contact with the thrust sheets 723, wherein one surface of the thrust sheet and the surface of the boss 7121 are movable relative to each other, and together form a bearing model.

[0068] Referring to FIG. 16, optionally, the boss 7121 is uniformly provided with a plurality of first radial oil grooves 7122 along a radial direction, and length directions of the first radial oil grooves 7122 are in radial directions of the thrust sheets 723. Optionally, four first radial oil grooves 7122 may be provided.

[0069] All of the roller bushing 722, the thrust sheets 723 and the roller 721 use an interference fit, so as to reduce the moving surface, and increase the moving speed of the friction surface. According to the dynamic pressure lubrication theory, within a certain range, the coefficient of friction decreases as the speed of relative movement of the friction surface increases. Therefore, increase of the speed of relative movement can enhance the dynamic pressure lubrication effect, so that a thicker dynamic pressure lubricant film is formed on the corresponding friction surface to avoid solid contact, thereby reducing the coefficient of friction and wear.

[0070] By providing the communication hole 713, the following effects can be obtained: (1) when flowing downwards from the communication hole 713, the lubricating oil above the guide piston 71 is uniformly distributed right at middle of and right above the second mounting hole of the roller 721, the lubricating oil is uniformly distributed on the bus bar of the roller 721, and the distribution of the lubricating oil on the surface of the roller 721 is not affected by the forward and reverse rotation (i.e. the lubricating oil can be uniformly distributed on the surface of the roller 721 no matter the roller 721 rotates forward or reversely); (2) the vertical force distribution of the guide piston 71 is improved, i.e. the pressure of the plunger 43 is distributed to a thicker portion around the communication hole 713, so as to equalize the overall stress, reduce the maximum stress, and improve the reliability of the system bearing capacity. In the guide piston in the prior art, the communication hole is provided around the center of the guide piston, and the center portion (i.e. portion wherein the communication hole of the present application is located) is solid, in this case, the portion around the communication hole is thin and subjected to a relatively large stress; and (3) in the pump assembly, when the guide piston 71 operates with the outer spring seat 61, the lubricating oil outlet passage 615 of the outer spring seat 61 communicates with the plunger couple 4, so that the lubricating oil leaked above the plunger couple 4 can be discharged through the lubricating oil outlet passage, thereby the first plunger spring 51 can be prevented from blocking the oil hole, and the actual flow area of the lubricating oil is increased.

[0071] Optionally, the annular groove 7211 is formed by processing after finishing the inner hole and an outer periphery of the roller 721. As in FIG. 20a and FIG. 20b, the arrangement of the annular groove 7211 reduces stiffness at two ends of the roller 721, and when the surface of the roller 721 is subjected to radial pressure, the roller 721 near the annular groove 7211 can be automatically deformed, meanwhile, after the annular groove 7211 is processed, the outer periphery and the inner hole of the roller 721 automatically collapse, microscopic arc surfaces are formed at two ends of the inner hole and the outer periphery of the roller 721, then reducing the geometric stress concentration at two ends of the roller 721, and further, the surface of the roller 721 is subjected to a uniform force. In the above, the geometric stress concentration herein refers to that when the surface of the roller 721 is stressed, the contact stresses at two ends of the bus bar of the roller 721 are obviously greater than the contact stress in the middle of the bus bar. A groove wall of the annular groove 7211 is formed in an arc shape, which can effectively reduce the geometric stress concentration present on an outer cylindrical surface of the roller 721 and the side compression effect of the inner hole of the roller 721 during the rotation of the roller 721, so that the inner and outer working surface stresses of the roller assembly 72 are distributed uniformly, thereby reducing the probability of seizing between the roller assembly 72 and the roller pin 73. In the above, the side compression effect described above specifically refers to a phenomenon that when the shaft cooperates with the hole, since there must be a certain angle error between the shaft and the hole during operation, that is, axes of the shaft and the hole are not parallel to each other, it in turn causes that the shaft and the hole are closer to each other at one side and farther from each other at the other side during operation, thereby resulting in that the closer sides are stressed greatly, and the farther sides are stressed less.

[0072] The boss 7121 forms a thrust bearing model with the corresponding friction surface (the end surface of the roller assembly 72). That is, the first radial oil groove 7122 is filled up with lubricating oil, and provides sufficient lubrication for the moving surface (the end surface of the roller assembly 72), a dynamic pressure lubricant film is formed on the end surface of the roller 721 by using the moving speed of the end surface of the roller 721, to separate the boss 7121 of the guide piston 71 from the end surface of the roller assembly 72, reduce wear, and reduce the coefficient of friction. The first radial oil groove 7122 is provided on the boss 7121 of the guide piston 71. Compared with the case where the first radial oil groove 7122 is provided on the roller assembly 72, the guide piston 71 does not rotate relatively, the high- and low-pressure lubricant film zones on the friction surface are distributed relatively stationary, and the axial direction of the roller assembly 72 thereby is relatively stationary. If the first radial oil groove is provided on a moving part (the end surface of the roller assembly 72), relative movement of the lubricant film distribution will be caused due to the relative movement of the first radial oil groove 7122 with respect to the guide piston 71, which in turn results in excessive additional axial vibrations of the roller 721 and reduces the overall dynamic performance of the roller assembly 72.

[0073] For the roller 721, the roller 721 adopts the design of end grooving and deformation so as to reduce boundary stress; specifically, for the roller 721, during processing thereof, the outer periphery of the roller 721 and the inner hole of the roller 721 are ground first, then the annular grooves 7211 at two axial end surfaces are grooved and processed, and after the grooving process is completed, the bus bars of the outer periphery and the inner hole of the roller 721 are naturally deformed into an arc, which can effectively weaken the geometric stress concentration present on the outer cylindrical surface of the roller and the side compression effect of the inner hole of the roller 721 during rotation of the roller 721, so that the inner and outer working surface stresses of the roller assembly 72 are distributed uniformly, thus reducing the probability of seizing between the roller assembly 72 and the roller pin 73. The manner of interference fit between the roller bushing 722 and the roller 721 increases the relative speed between the moving surface of the roller bushing 722 and the roller pin 73, and the end surface of the roller bushing 722 moves at a high speed to form an effective dynamic pressure lubricant film between the roller bushing and the roller pin 73, thereby improving the dynamic pressure lubricating effect, and reducing the probability of seizing between the roller bushing 722 and the roller pin 73; the manner of interference fit adopted between the roller 721 and the thrust sheet 723 increases the relative speed between the moving surface of the thrust sheet 723 and the boss 7121, the end surface of the thrust sheet 723 moves at a high speed to form an effective dynamic pressure lubricant film between the thrust sheet and the boss 7121, thereby preventing the boss 7121 from being fitted/attached tightly to the thrust sheet 723, and avoiding excessive wear caused by insufficient oil supply on the end surface of the thrust sheet 723. The formation of the dynamic pressure lubricant film can improve the dynamic pressure lubricating effect, and reduce the possibility of seizing between the thrust sheets 723 and the boss 7121.

[0074] Preferably optionally, as shown in FIG. 17, the outer surface of the roller pin 73 is provided as a cylindrical surface, and a first waist-shaped groove 731 and a second waist-shaped groove 732 are provided at each of two positions on the cylindrical surface. The first waist-shaped groove 731 and the second waist-shaped groove 732 are provided at a middle of the roller pin 73, wherein the second waist-shaped groove 732 is provided in the first waist-shaped groove 731 and recessed with respect to the first waist-shaped groove 731, and the first waist-shaped groove and the second waist-shaped groove together form a stepped shape; the first waist-shaped groove 731 and the second waist-shaped groove 732 are both elongated, and there is a relatively large contact area between the lubricating oil in the first waist-shaped groove 731 and the second waist-shaped groove 732 with corresponding friction surfaces, and more lubricating oil is brought into the bearing surface to form the dynamic pressure lubricant film through full use of the moving speed of the corresponding moving surface, further forming a thicker lubricating oil film. Two sides of the first waist-shaped groove 731 are arranged in a waist shape, to reduce the stress concentration brought about by the grooving on the surface of the roller pin 73. The arrangement of the first waist-shaped groove 731 and the second waist-shaped groove 732 can increase the flow rate of surface lubricating oil.

[0075] Optionally, a small-angle wedge groove with an angle ranging 5-10 ° is formed between the first waist-shaped groove 731 located at an outer layer and an outer surface of the roller bushing 722, and a second oil hole 73 is provided in each waist-shaped groove 732 located at an inner layer; the two waist-shaped grooves have an included angle of 70-120 °(an actual value may be determined according to a simulation calculation result, and may be selected as 90 ° in an example) in a plane perpendicular to the axial direction of the roller pin, and are located right above a pressure-bearing region, so as to reduce the influence of providing the waist-shaped groove on the surface on the area of the pressure-bearing region under the condition of sufficiently supplying oil to the friction surface, and further result in a larger angle of the pressure-bearing region and a smaller average pressure of the lubricant film in the pressure-bearing region; the first waist-shaped groove 731 and the corresponding friction surface form a small-angle convergent wedge shape, to enhance the extrusion effect in the dynamic pressure lubrication; the second waist-shaped groove 732 is mainly used to store more lubricating oil, and guarantee sufficient oil supply to the friction surface, then the lubrication on the surface of the roller pin is not affected even if the oil supply is poor within a short period of time, and when the lubrication system is out of work, the probability of the system seizing is reduced; the two second oil holes 733 at the two positions communicate with each other through a lubricating oil outlet passage, and the two second oil holes 733 are arranged with 90 ° therebetween.

[0076] Optionally, as shown in FIG. 15 and FIG. 16, an outer surface of the guide piston 71 is provided as a cylindrical surface, on which a circumferential oil groove 715, a partial circumferential oil groove 714, a first axial oil groove 716 and a vertical groove 717 are provided, the vertical groove 717 is provided in the circumferential oil groove 715, and the vertical groove 717 communicates with the partial circumferential oil groove 714 through the first axial oil groove 716; both upper and lower edges of the circumferential oil groove 715 are each provided with a chamfer of 1~10 ° with both upper and lower ends of the guide piston 71, and the chamfer forms a small-angle convergence wedge shape with the corresponding moving surface during movement, thus enhancing the extrusion effect during dynamic pressure lubrication. The surface lubrication state of the guide piston 71 is improved, so that a thicker dynamic pressure lubricant film is established, the friction is reduced, and the probability of seizing is reduced. According to relevant data and experiments, when the chamfer is too large (for example, 45 ° or 90 °), the chamfer cannot improve lubrication, but instead has a scraping effect on the corresponding friction surface, i.e. scrapes off the surface lubricating oil, and then reduces the lubricating effect.

[0077] Referring to FIG. 14, the cylindrical surface of the guide piston 71 is further provided with an inclined hole 718, and two ends of the inclined hole 718 respectively communicates with the inner walls of the circumferential oil groove 715 and the second mounting hole 712.

[0078] Referring to FIG. 16, the cylindrical surface is further provided with a second axial oil groove 719 communicating with the circumferential oil groove 715.

[0079] Optionally, the cylindrical surface is further provided with a first straight hole 7100 and a second straight hole 7101 connected with each other, the first straight hole 7100 communicates with the first axial oil groove 716, and the second straight hole 7101 communicates with the first mounting hole 711; the first straight hole 7100 and the second straight hole 7101 supply lubricating oil to the lower spring seat assembly 6 inside the guide piston 71, thereby reducing the wear of the corresponding moving surface.

[0080] Referring to FIG. 13, the roller pin 73 is provided on an outer peripheral surface thereof with a lubricating oil inlet passage 735, the lubricating oil inlet passage 735 is provided opposite to the inclined hole 718, and the lubricating oil inlet passage 735 communicates with the lubricating oil outlet passage.

[0081] Optionally, the roller pin 73 is provided on the outer peripheral surface thereof with a DLC (Diamond-Like Carbon) coating; the DLC coating has the characteristics of high hardness, low coefficient of friction, wear resistance and high temperature resistance. When the lubrication between the roller bushing 722 and the roller pin 73 is poor, the friction pair formed by the DLC coating and the roller bushing 722 of a copper alloy bearing can still operate well, which can further reduce the probability of seizing between the roller pin 73 and the roller bushing 722.

[0082] Optionally, the roller bushing 722 is made of a copper alloy.

[0083] Optionally, the thrust sheets 723 are each made of a copper alloy.

[0084] Optionally, forced lubrication and dynamic pressure lubrication are adopted between the roller pin 723 and the roller bushing 722.

[0085] Optionally, forced lubrication and dynamic pressure lubrication are adopted between the thrust sheets 723 and the boss 7121.

[0086] It should be noted that the forced lubrication is a lubrication method in which the pressure of the lubricating oil is forcibly increased by an external force so as to establish a lubricating film between contact surfaces of various components. The dynamic pressure lubrication is a lubrication method in which a dynamic pressure lubricant film is formed by bringing the lubricating oil into the friction surface through the movement of the moving surface. At normal and low speeds, the parts are forcibly lubricated mainly by the pressure generated by the lubricating oil pump, and at a high speed, the parts are lubricated mainly by the dynamic pressure lubricant film generated by the movement of the parts.

[0087] Optionally, the roller bushing 722 and the thrust sheets 723 are made of a bronze alloy, and due to the characteristics of low coefficient of friction, good wear resistance, self-lubrication property and impact resistance of the bronze alloy, the friction property shown when there is solid friction between the inner hole and the end surface of the roller assembly 72 and the corresponding moving surfaces is improved, thus the coefficient of friction is reduced, the impact resistance is improved, and the bearing capability is improved.

[0088] Optionally, as shown in FIG. 13, FIG. 14, FIG. 16 and FIG. 18, a first chamfer 7102 is disposed at an upper end surface outer periphery and a lower end surface outer periphery of the guide piston 71 and the circumferential annular groove 715.

[0089] The hole wall of the first mounting hole 711 is provided with a second chamfer 7110.

[0090] Optionally, the hole wall of the second mounting hole 712 is provided with a fourth hole 7123.

[0091] A fifth hole 736 is provided on the outer peripheral surface of the roller pin 73;

[0092] A return spring 737 and a stop pin 738 are disposed in sequence in the fifth hole 736, and the stop pin 738 partially extends into the fourth hole 7123. The fifth hole 736 is configured to mount the stop pin 738, make the roller pin 73 and the guide piston 71 relatively stationary, reduce the number of relative moving surfaces of the roller pin 73 and the roller 721, further increase the speed of the relative moving surfaces, and further enhance the dynamic pressure lubricating effect, of which the principle is the same as the interference fit between the roller and the bushing. Optionally, when the roller assembly 72 and the roller pin 73 are assembled onto the guide piston 71, the roller bushing 722 and the thrust sheet 723 are firstly mounted on the roller 721 in a cold mounting manner; then the return spring 737 and the stop pin 738 are placed in sequence into the fifth hole 736 of the roller pin 73; next, the roller assembly 72 is placed at a lower portion of the guide piston 71, and the roller pin 73 sequentially passes through one side of the fourth hole 7123 at the lower end of the guide piston 71, the inner hole of the roller assembly 72 (specifically, the inner hole of the roller bushing), and the other side of the fourth hole 7123 at the lower end of the guide piston 71; then, the stop pin 738 is pressed with hand so that its height is lower than the second mounting hole 712, meanwhile, the roller pin 73 is pushed until the stop pin 738 is sprung into the fourth hole 7123 of the guide piston 71 under the action of the return spring 737. That is to say, the lubricating oil flowing out of the pump body 1 of an oil injection pump flows into the circumferential oil groove 715 through the second axial oil groove 719, then flows into the partial circumferential oil groove 714 through the vertical groove 717 and the first axial oil groove 716, thus realizing the lubrication between the guide piston 71 and the pump body 1 of the oil injection pump, and as there is a quite small gap between the guide piston 71 and the center hole of the pump body 1 cooperating with the guide piston, the lubricating oil entering the second axial oil groove 719 and the circumferential oil groove 715 of the guide piston 71 maintains a certain pressure, and a lubricating oil film may be formed between the outer periphery of the guide piston 71 and the center hole of the pump body 1; meanwhile, the lubricating oil in the second axial oil groove 719 partially flows into the lubricating oil inlet passage 735 through the inclined hole 718, so as to go deep into the inside of the roller pin 73, then flows out through the second oil hole 733 to the outer peripheral surface of the roller pin 73, to carry out deep lubrication between the roller pin 73 and the roller bushing 722, and forms a lubricating oil film between the roller pin 73 and the roller bushing 722.

[0093] Optionally, the angles of the first chamfer 7102 and the second chamfer 7110 range 1-10 °, and when the guide piston 71 cooperates with the center hole on the pump body 1, a small-angle convergence wedge shape can be formed, thus enhancing the extrusion effect in the dynamic pressure lubrication, increasing the thickness of the lubricant film on the surface of the guide piston 71 during operation, and thereby reducing the probability of seizing between the guide piston 71 and the pump body 1.

[0094] Optionally, as shown in FIG. 18 and FIG. 19, the lubricating oil inlet passage 735 includes a third radial oil passage 7351 disposed along a radial direction of the roller pin 73 and an axial oil passage 7352 disposed along an axial direction of the roller pin 73, and the third radial oil passage 7351 is connected to the axial oil passage 7352; and the axial oil passage 7352 is connected to the second oil hole 733 in the second waist-shaped groove 732.

[0095] The above embodiments only describe one or more embodiments of the present disclosure, but those ordinarily skilled in the art should understand that the present disclosure can be implemented in many other forms without departing from the spirit and scope thereof. Therefore, the examples and embodiments illustrated are to be considered as illustrative and nonrestrictive, and various modifications and substitutions may be encompassed by the present disclosure without departing from the spirit and scope of the present disclosure as defined by various appended claims.

Claims

1. An electrically-controlled monolithic high-pressure oil pump for a marine low-speed engine, **characterized by** comprising:

    a pump body (1), wherein the pump body (1) is provided with a center hole (10) along an axial direction;
    a pump cover (2), wherein the pump cover (2) is mounted on an upper end surface of the pump body (1);
    an oil inlet-outlet valve assembly (3), a plunger couple (4), a plunger spring (5), a lower spring seat assembly (6) and a guide piston assembly (7), all of which are assembled in the center hole of the pump body (1); and
    an electrically-controlled proportional valve (8), wherein the electrically-controlled proportional valve (8) is assembled on a side surface of the pump body (1),
    wherein the oil inlet-outlet valve assembly (3) comprises: an oil inlet valve assembly (31) and an oil outlet valve assembly (32);
    the oil inlet valve assembly (31) comprises: an oil inlet valve seat (311), an oil inlet valve (312) and an oil inlet valve spring (313);
    the oil inlet valve (312) is mounted in a center hole of the oil inlet valve seat (311), the oil inlet valve spring (313) is restrained between the oil inlet valve (312) and a bore wall of the oil inlet valve seat (311), and the oil inlet valve (312) is configured to form a conical seal with the oil inlet valve seat (311) under compression of the oil inlet valve spring (313);
    the oil outlet valve assembly (32) comprises: an oil outlet valve seat (321), an oil outlet valve (322), an oil outlet valve spring (323) and an oil outlet valve spring seat (324);
    the oil outlet valve spring seat (324) is mounted on an upper end of the oil outlet valve seat (321), the oil outlet valve (322) is mounted in a center hole of the oil outlet valve seat (321), the oil outlet valve spring (323) is restrained between the oil outlet valve (322) and the oil outlet valve spring seat (324), and the oil outlet valve (322) is configured to form a conical seal with the oil outlet valve seat (321) under compression of the oil outlet valve spring (323);
    a high-pressure oil outlet chamber (33) is formed between the oil outlet valve seat (321) and the oil inlet valve seat (311);
    a high-pressure oil chamber (41) is formed in the plunger couple (4), and the high-pressure oil chamber (41) communicates with the high-pressure oil outlet chamber (33) through a first oil hole (13) of the oil inlet valve on the oil inlet valve seat (311);
    the electrically-controlled proportional valve (8) communicates with an oil inlet hole of the oil inlet valve seat (311) through the first oil hole (13) on the pump body (1), and the oil inlet hole of the oil inlet valve seat is configured to communicate with or be disconnected from the high-pressure oil chamber (41); and
    the electrically-controlled proportional valve (8) is provided thereon with a cooling circulation oil passage, wherein cooling oil from a cooling oil passage of the pump body (1), after being injected into the cooling circulation oil passage, is returned to the cooling oil passage of the pump body (1).

2. The oil pump according to claim 1, wherein the plunger couple (4) comprises:

    a plunger sleeve (42), wherein the plunger sleeve (42) is disposed at a lower end of the oil inlet valve seat (311); and
    a plunger (43), wherein the plunger (43) is slidably inserted into a center hole of the plunger sleeve (42), and the high-pressure oil chamber (41) is defined by the plunger sleeve (42), the plunger (43) and the oil inlet valve seat (311),
    wherein an inner wall of the plunger sleeve (42) is provided with a first annular groove (421) and a second annular groove (422);
    the pump body (1) is provided with a mixed oil outlet passage and a lubricating oil supply passage (12), wherein the mixed oil outlet passage communicates with the first annular groove (421) through a mixed oil passage (423) on the plunger sleeve (42), and the lubricating oil supply passage (12) communicates with the second annular groove (422) through the lubricating oil passage (424) on the plunger sleeve (42); and the first annular groove (421) is located above the second annular groove (422).

3. The oil pump according to claim 2, wherein the lower spring seat assembly (6) is disposed below the plunger couple (4), and the lower spring seat assembly (6) comprises:

an outer spring seat (61), wherein the outer spring seat (61) is as a whole of a boss type structure having a central part thick and an outer side thin, and an upper end surface of the outer spring seat (61) is provided with a counterbore (611) having a concave spherical surface;
an upper sphere (62), wherein a lower portion of the upper sphere (62) is mounted in the counterbore (611), and a lower end surface of the upper sphere (62) is provided with a convex spherical surface matched with the concave spherical surface; and
an inner spring seat (63), wherein the inner spring seat (63) is sheathed on an upper portion of the upper sphere (62), and the inner spring seat (63) has an axial through hole (631) penetrating upper and lower end surfaces,
wherein a lower cylindrical head (431) of the plunger (43) is restrained in the axial through hole (631), and a lower end surface of the lower cylindrical head (431) of the plunger (43) abuts against an upper end surface of the upper sphere (62).

4. The oil pump according to claim 3, wherein a spherical hole (612) is provided at a center of the counterbore (611), a third annular groove (613) is provided on a lower end surface of the outer spring seat (61), and the spherical hole (612) and the third annular groove (613) communicate with each other through a lubricating oil inlet passage (614);
an outer surface of the outer spring seat (61) is a conical surface, the conical surface is provided with a lubricating oil outlet passage (615), and the lubricating oil outlet passage (615) communicates with a lower end surface of the outer spring seat (61), and the lubricating oil outlet passage (615) is arranged obliquely;
the upper sphere (62) is provided with a circumferential annular groove (621) in a circumferential direction thereof;
a positioning screw (64) is mounted in the circumferential annular groove (621) after passing through a positioning screw hole (616) of the outer spring seat (61); and
a distance between an upper surface and a lower surface of the circumferential annular groove (621) is greater than a cylindrical diameter of a portion of the positioning screw (64) located in the circumferential annular groove (621).

5. The oil pump according to claim 4, wherein eight lubricating oil outlet passages (615) are provided, and the eight lubricating oil outlet passages (615) respectively communicate with a bottom end surface of the outer spring seat (61).

6. The oil pump according to any one of claims 3 to 5, wherein the axial through hole (631) provided inside the inner spring seat (63) comprises a first hole (6311), a second hole (6312) and a third hole (6313) that have diameters respectively, wherein one diameter is less than another in sequence from top to bottom,
wherein a first guide hole (6314) having a gradually increasing diameter is provided between the second hole (6312) and the third hole (6313);
one side of the third hole (6313) facing the upper sphere (62) is provided with a second guide hole (6315) having a gradually increasing diameter;
hole walls of the first guide hole (6314) and the second guide hole (6315) are formed as guide conical surfaces (6316);
a part of an upper portion of the upper sphere (62) penetrating the second guide hole (6315) is located in the third hole (6313);
the upper sphere (62) and the third hole (6313) have a gap therebetween which is greater than or equal to 1 mm, and
the counterbore (611) and the upper sphere (62) have a gap therebetween which is greater than or equal to 1 mm.

7. The oil pump according to claim 6, wherein an outer peripheral surface of the inner spring seat (63) and a hole wall of the second hole (6312) are each provided with a relief groove (6317); and an upper end surface of the inner spring seat (63) is provided with a weight-reduction annular groove (6318).

8. The oil pump according to any one of claims 4 to 7, further comprising:

an upper spring seat (9), which is sheathed on the plunger sleeve (42), and located at an upper end of the inner spring seat (63); and
the plunger spring (5) comprises:

a first plunger spring (51), wherein the first plunger spring (51) is press-fitted between the upper spring seat (9) and the outer spring seat (61), and
a second plunger spring (52), wherein the second plunger spring (52) is press-fitted between the upper spring seat (9) and the inner spring seat (63).

9. The oil pump according to claim 8, wherein a diameter of the concave spherical surface in the outer spring seat (61) and a diameter of the convex spherical surface of the upper sphere (62) are respectively 20 to 100 times a diameter of the plunger (43).

10. The oil pump according to any one of claims 1 to 9, wherein the guide piston assembly (7) comprises:

a guide piston (71), wherein the guide piston is provided with a first mounting hole (711) at a central position of an upper end surface thereof, and a second mounting hole (712) on a lower end surface thereof, the first mounting hole (711) and the second mounting hole (712) communicate with each other through a communication hole (713), and the lower spring seat assembly (6) is mounted in the first mounting hole (711);
a roller assembly (72), comprising a roller (721) mounted in the second mounting hole (712), a roller bushing (722) interference-assembled in the roller (721), and a thrust sheet (723) interference-assembled at two axial ends of the roller (721), wherein an annular groove (7211) is provided in an axial direction of the roller (721), and a circular arc transition connection is formed between a groove bottom of the annular groove (7211) and an axial end surface of the roller (721); and
a roller pin (73), which is clearance-assembled in the roller bushing (722),
wherein a hole wall of the second mounting hole (712) is provided with a boss (7121), and the boss (7121) is in contact with the thrust sheets (723); and
the boss (7121) is uniformly provided with a plurality of first radial oil grooves (7122) along a radial direction, and length directions of the first radial oil grooves (7122) are in radial directions of the thrust sheet (723).

11. The oil pump according to claim 10, wherein an outer surface of the roller pin (73) is provided as a cylindrical surface, and a first waist-shaped groove (731) and a second waist-shaped groove (732) are provided at each of two positions on the cylindrical surface, the first waist-shaped groove (731) and the second waist-shaped groove (732) are provided at a middle of the roller pin (73);
a small-angle wedge groove with an angle between 5 ° and 20 ° is formed between the first waist-shaped groove (731) and an outer surface of the roller bushing (722), and the second waist-shaped groove (732) is provided therein with a second oil hole (733); and two second oil holes (733) at the two positions communicate with each other through a lubricating oil outlet passage, and the two second oil holes (733) are arranged with 70-120 ° therebetween.

12. The oil pump according to claim 11, wherein an outer surface of the guide piston (71) is provided as a cylindrical surface, on which a partial circumferential oil groove (714), a circumferential oil groove (715), a first axial oil groove (716) and a vertical groove (717) are provided, the vertical groove (717) is provided in the circumferential oil groove (715), and the vertical groove (717) communicates with the partial circumferential oil groove (714) through the first axial oil groove (716);
the cylindrical surface is further provided with an inclined hole (718), and two ends of the inclined hole (718) respectively communicate with inner walls of the circumferential oil groove (715) and the second mounting hole (712);
the cylindrical surface is further provided with a second axial oil groove (719) communicating with the circumferential oil groove (715);
the cylindrical surface is further provided with a first straight hole (7100) and a second straight hole (7101) connected with each other, the first straight hole (7100) communicates with the first axial oil groove (716), and the second straight hole (7101) communicates with the first mounting hole (711); and the roller pin (73) is provided on an outer peripheral surface thereof with a lubricating oil inlet passage (735), the lubricating oil inlet passage (735) is provided opposite to the inclined hole (718), and the lubricating oil inlet passage (735) communicates with the lubricating oil outlet passage.

13. The oil pump according to claim 12, wherein the lubricating oil inlet passage (735) comprises a third radial oil passage (7351) disposed along a radial direction of the roller pin (73) and an axial oil passage (7352) disposed along an axial direction of the roller pin (73), wherein the third radial oil passage (7351) is connected to the axial oil passage (7352); and the axial oil passage (7352) is connected to the second oil hole (733) in the second waist-shaped groove (732).

14. The oil pump according to any one of claims 10 to 13, wherein an outer peripheral surface of the roller pin (73) is provided with a diamond-like carbon (DLC) coating;
the roller bushing (722) is made of a copper alloy;
the thrust sheets (723) each are made of a copper alloy;
forced lubrication and dynamic pressure lubrication are used between the roller pin (723) and the roller bushing (722); and
forced lubrication and dynamic pressure lubrication are used between the thrust sheets (723) and the boss (7121).

15. The oil pump according to claim 14, wherein each of an upper end surface outer periphery and a lower end surface outer periphery of the guide piston (71) and the circumferential annular groove (715) is each provided with a first chamfer (7102);
the hole wall of the first mounting hole (711) is provided with a second chamfer (7110);
the hole wall of the second mounting hole (712) is provided with a fourth hole (7123);
a fifth hole (736) is provided on the outer peripheral

surface of the roller pin (73); and

a return spring (737) and a stop pin (738) are disposed in sequence in the fifth hole (736), and the stop pin (738) partially extends into the fourth hole (7123).

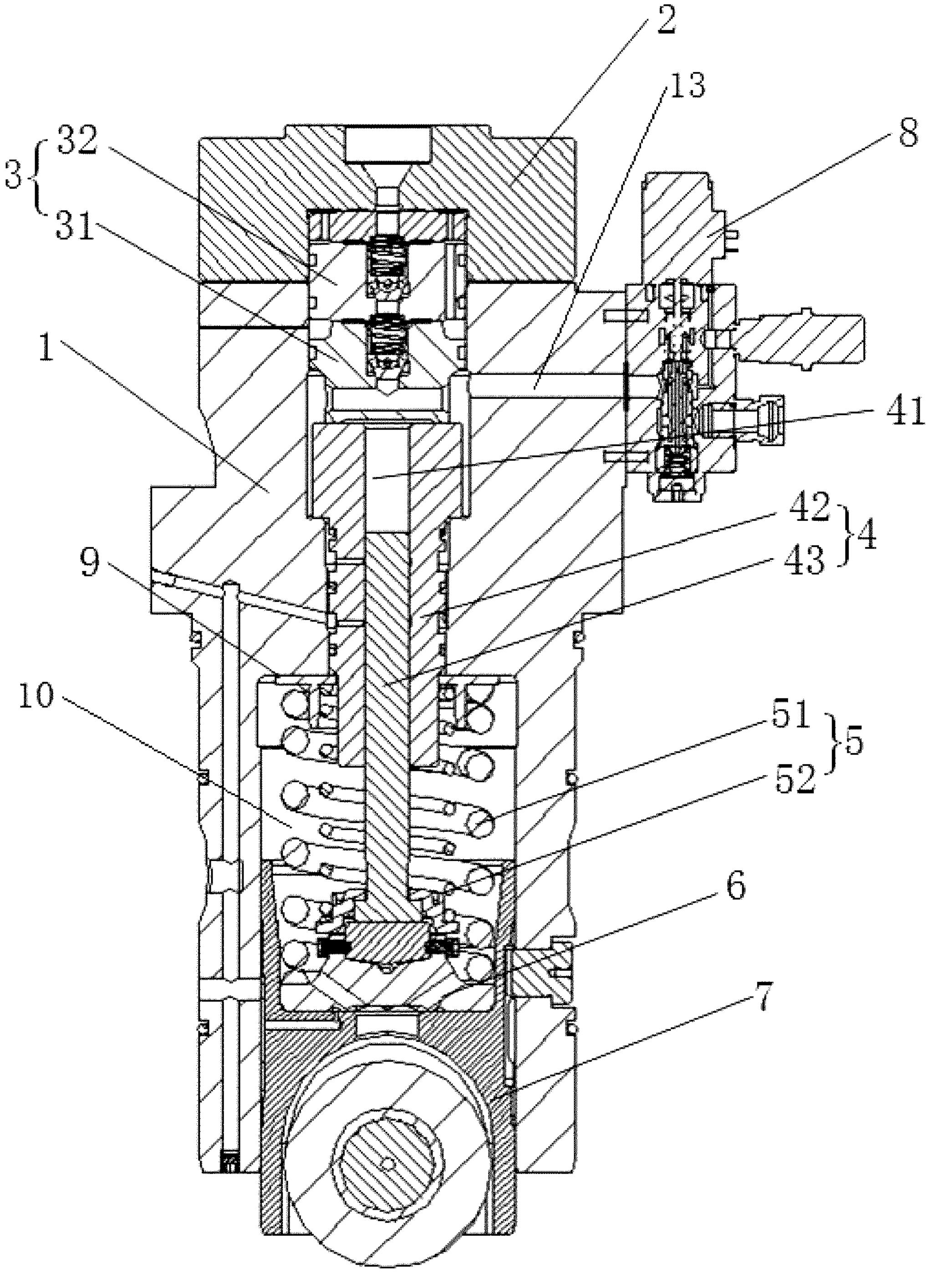

FIG.1

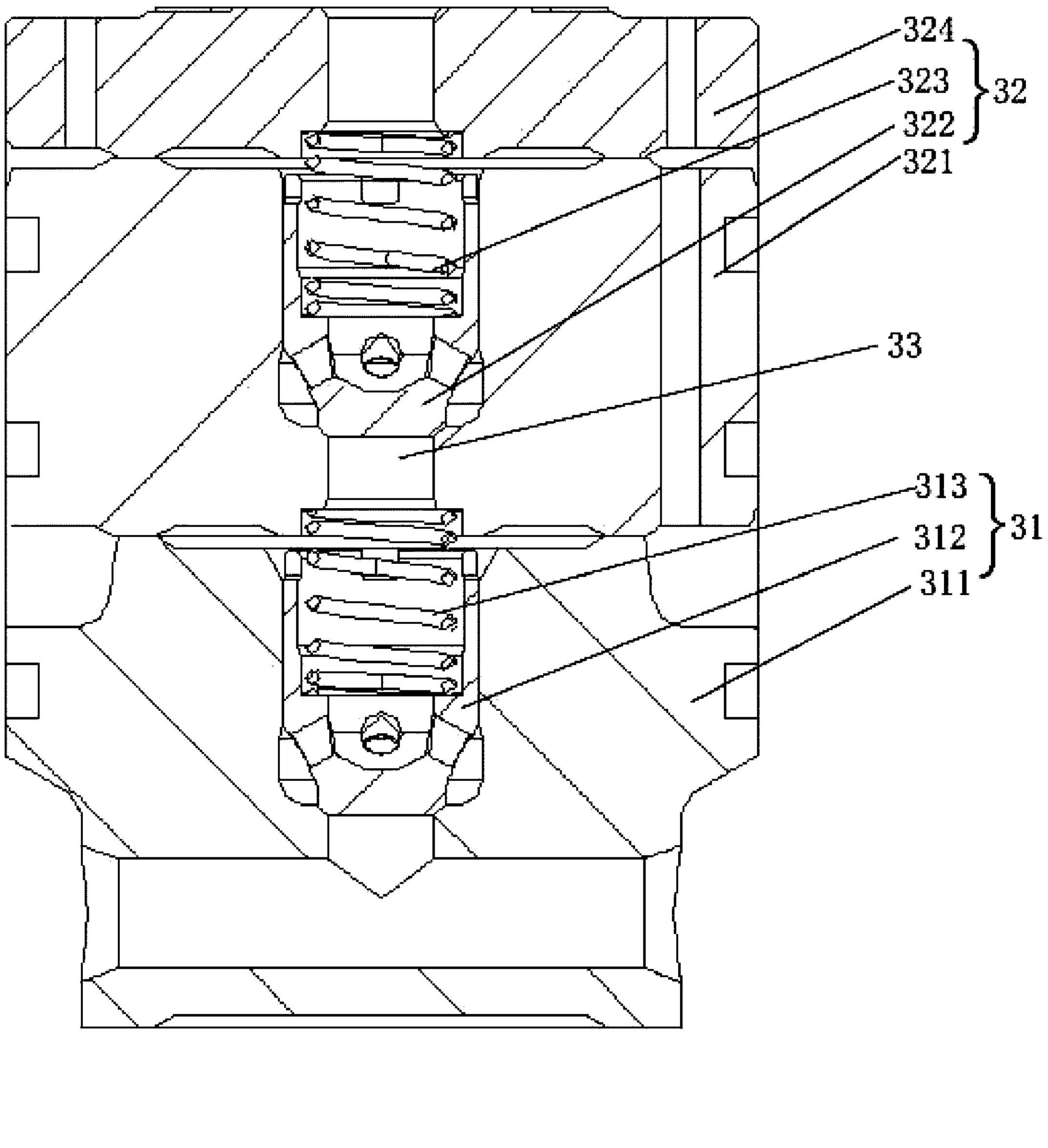

FIG.2

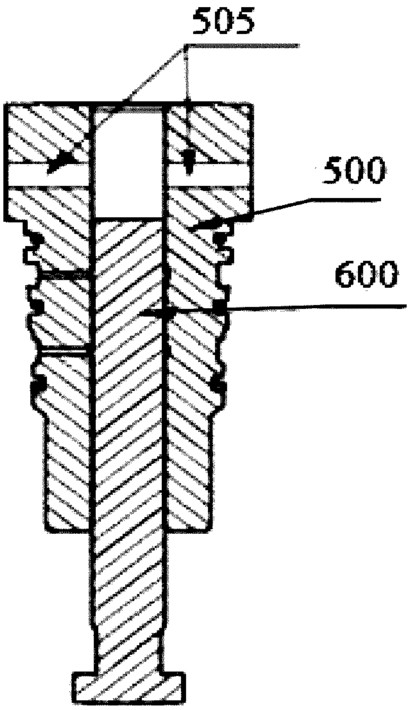

FIG.3

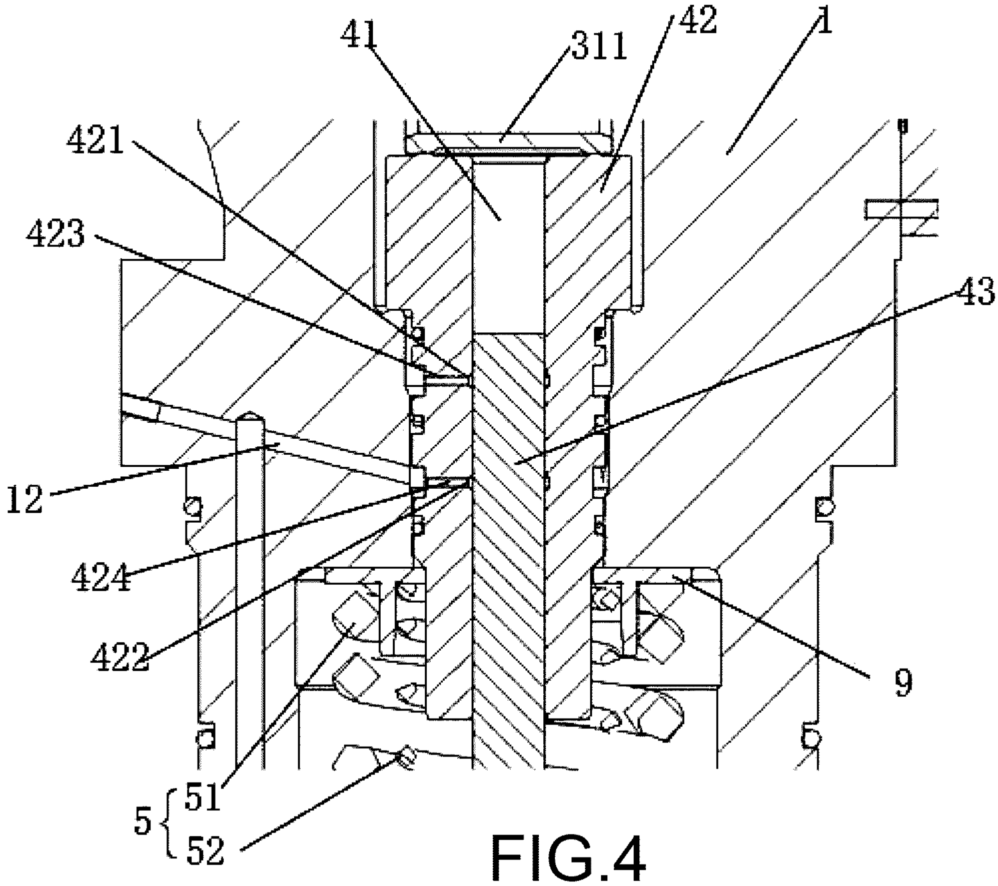

FIG.4

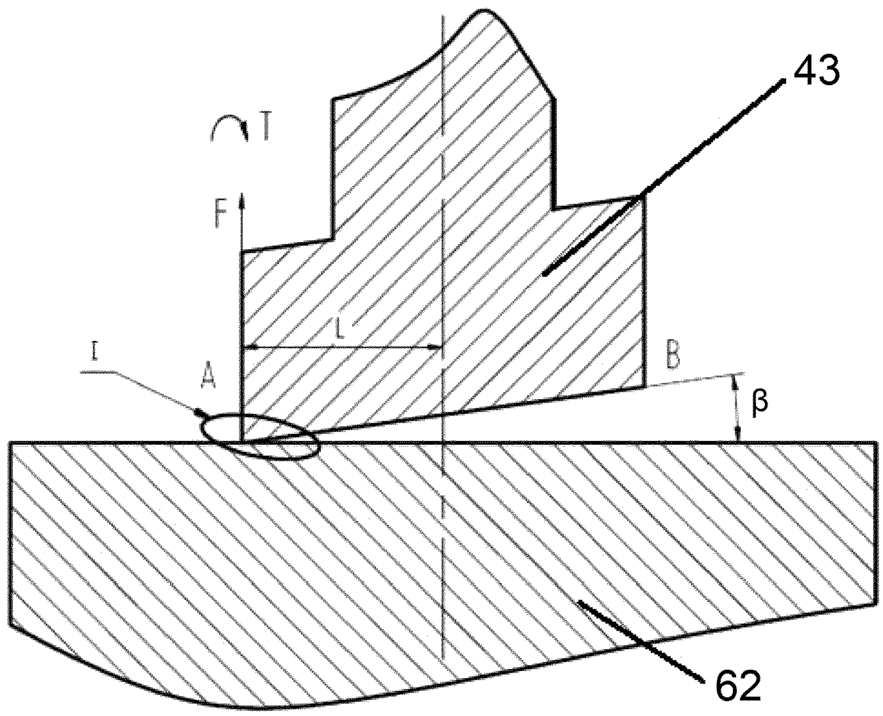

FIG.5a

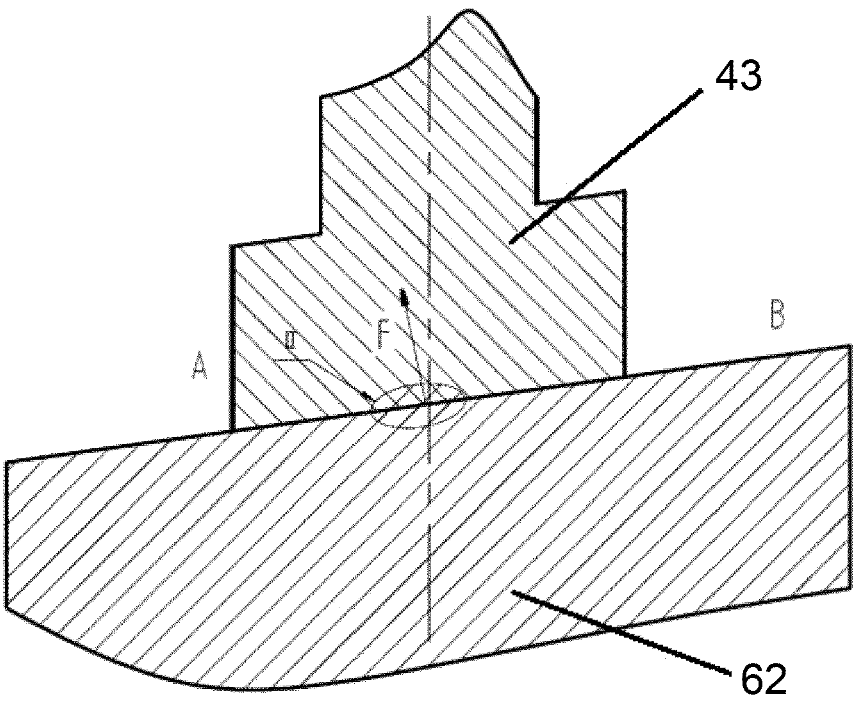

FIG.5b

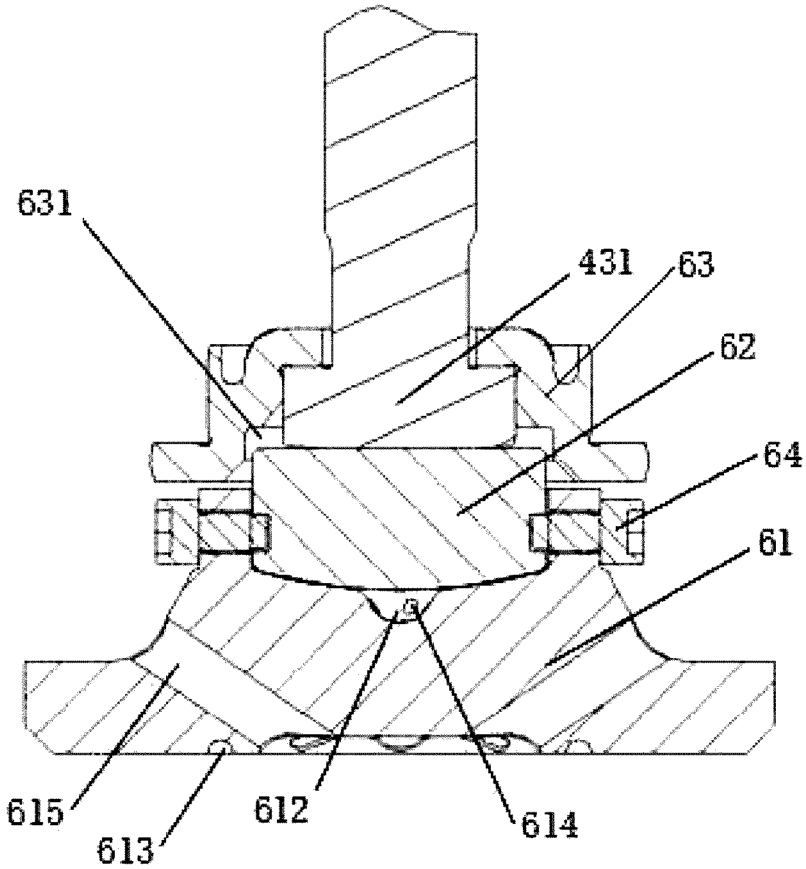
631
431
63
62
64
61
615
613
612
614

FIG.6

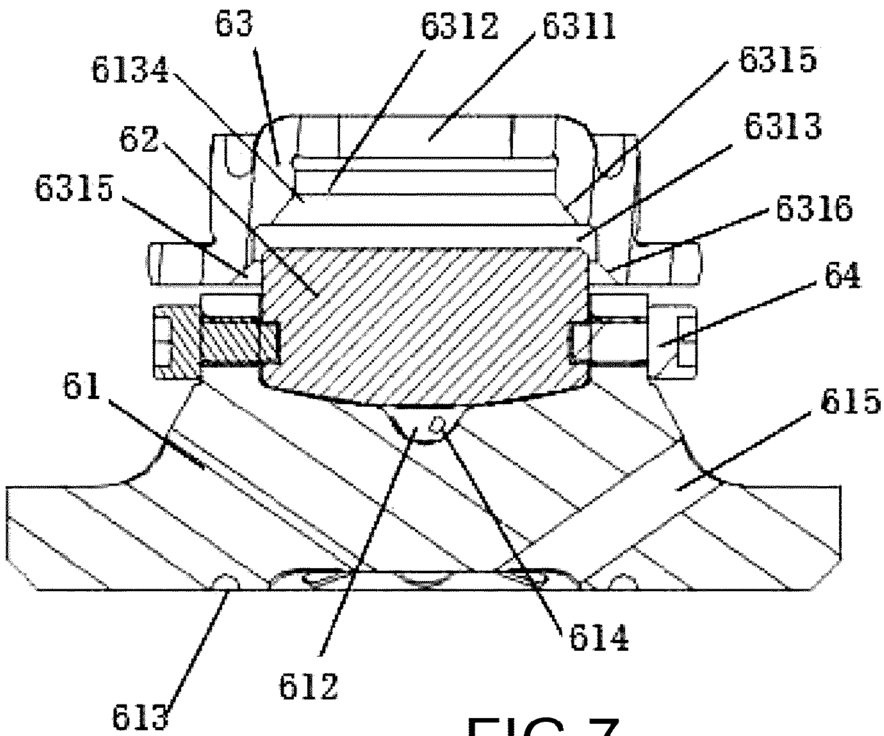
63
6312
6311
6134
6315
62
6313
6315
6316
64
61
615
614
612
613

FIG.7

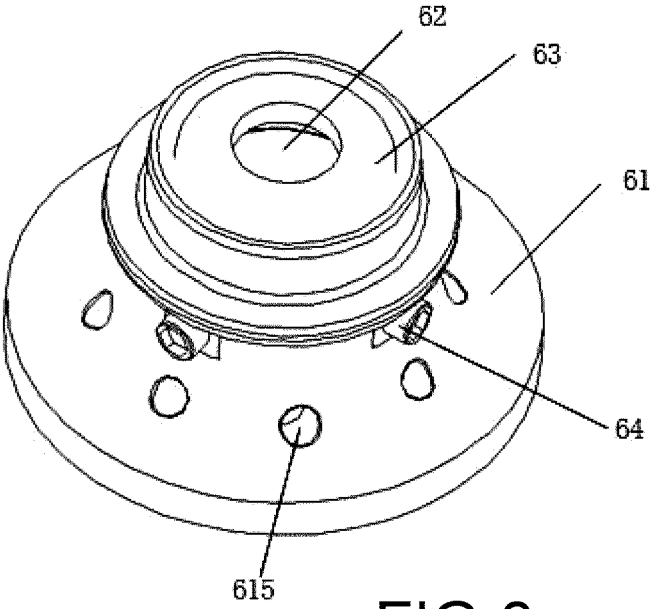

FIG.8

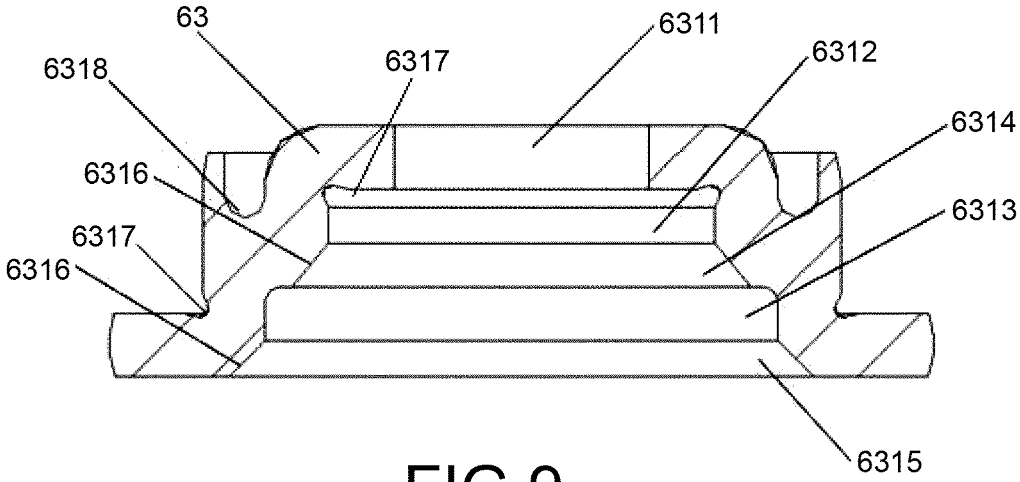

FIG.9

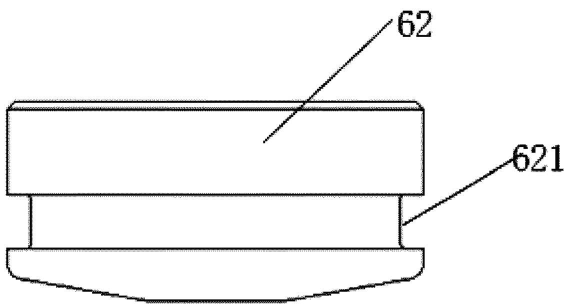

FIG.10

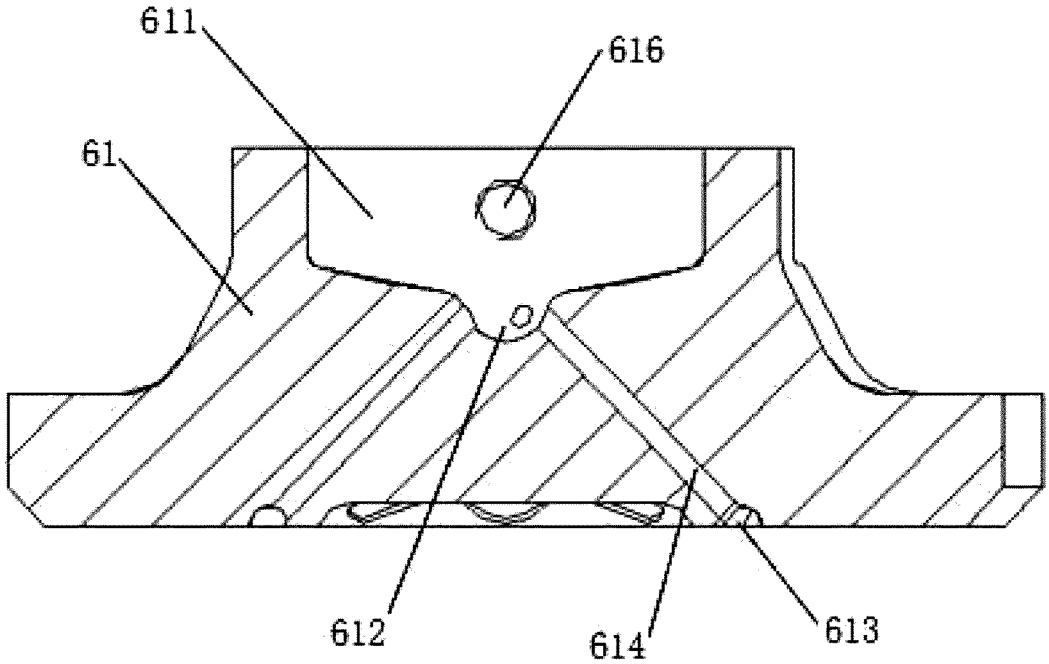

FIG.11

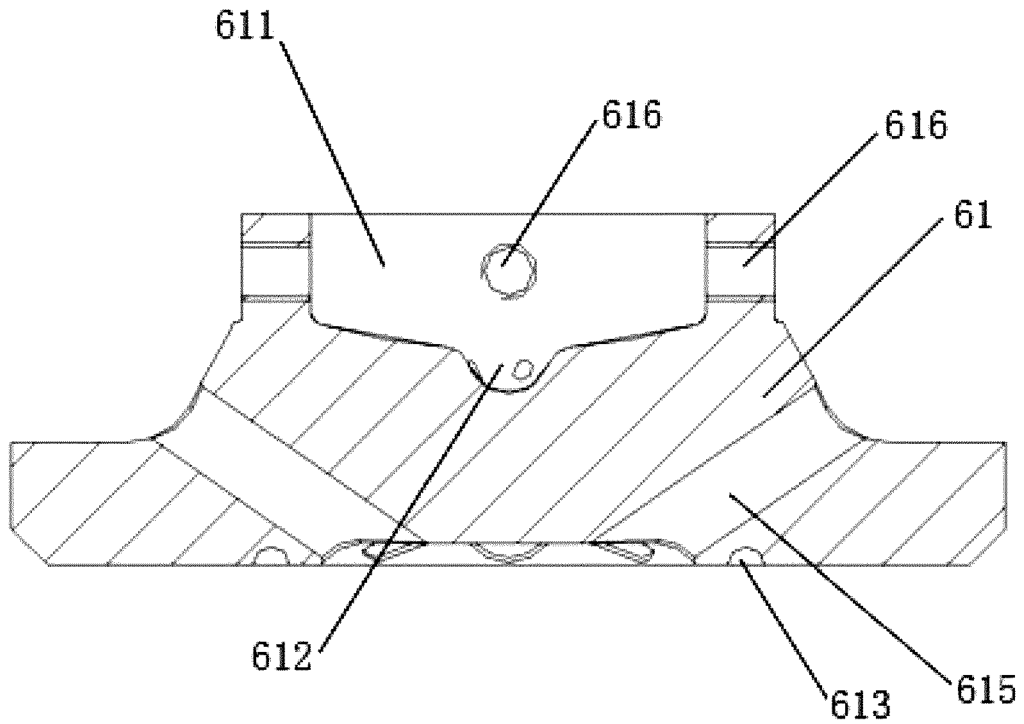

FIG.12

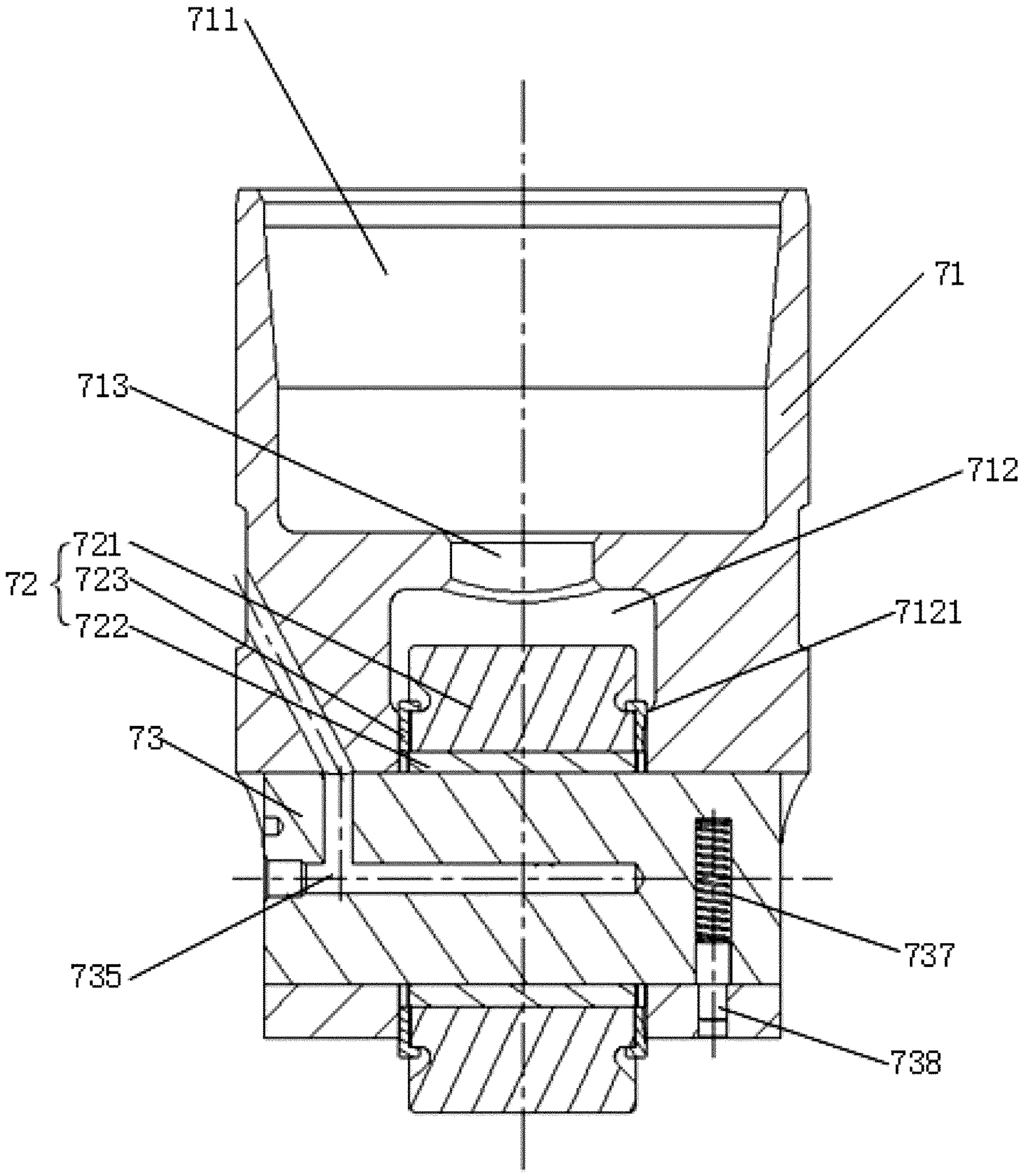

FIG.13

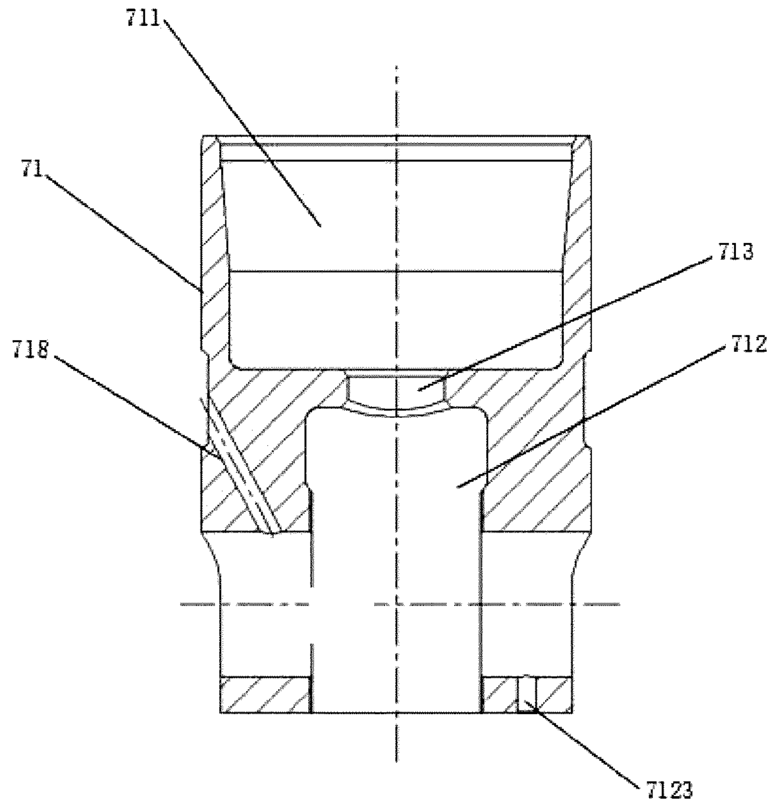

FIG.14

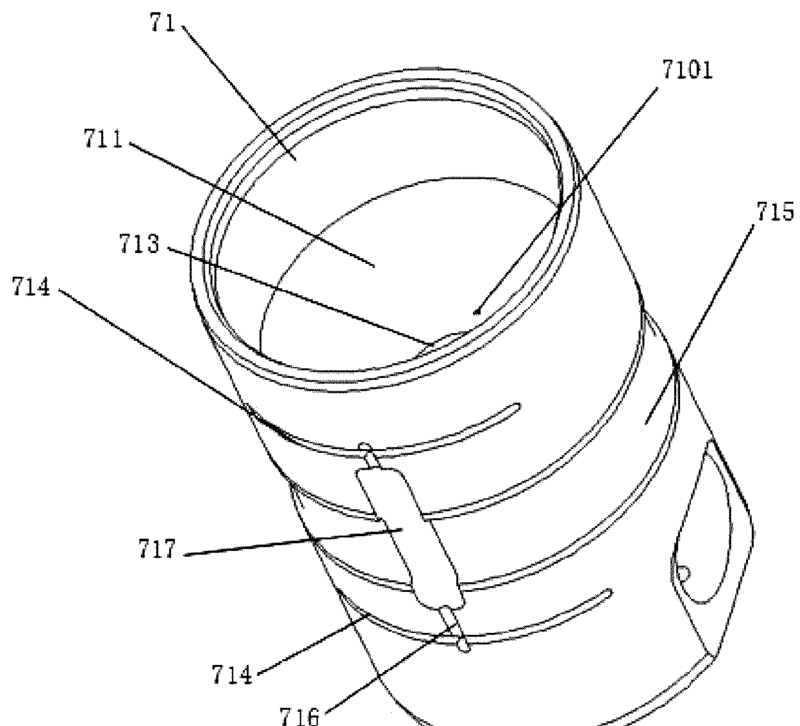

FIG.15

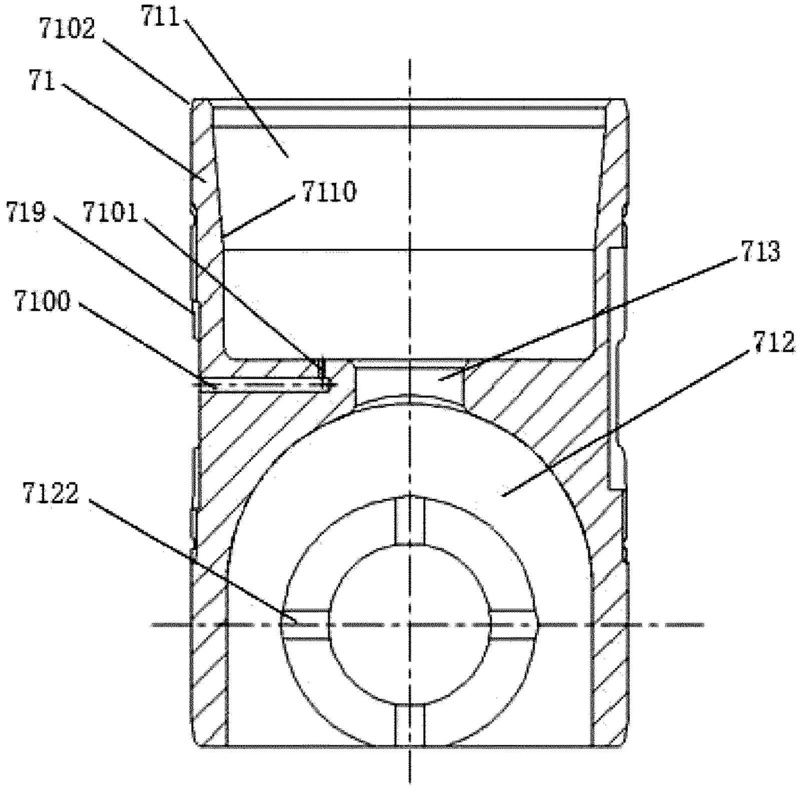
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FIG.16

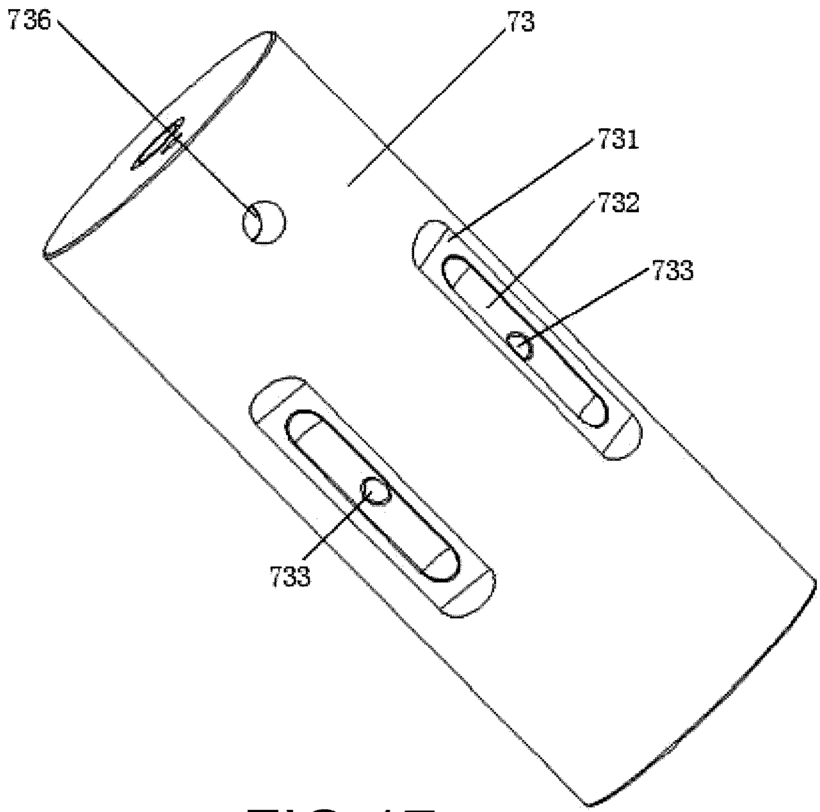
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FIG.17

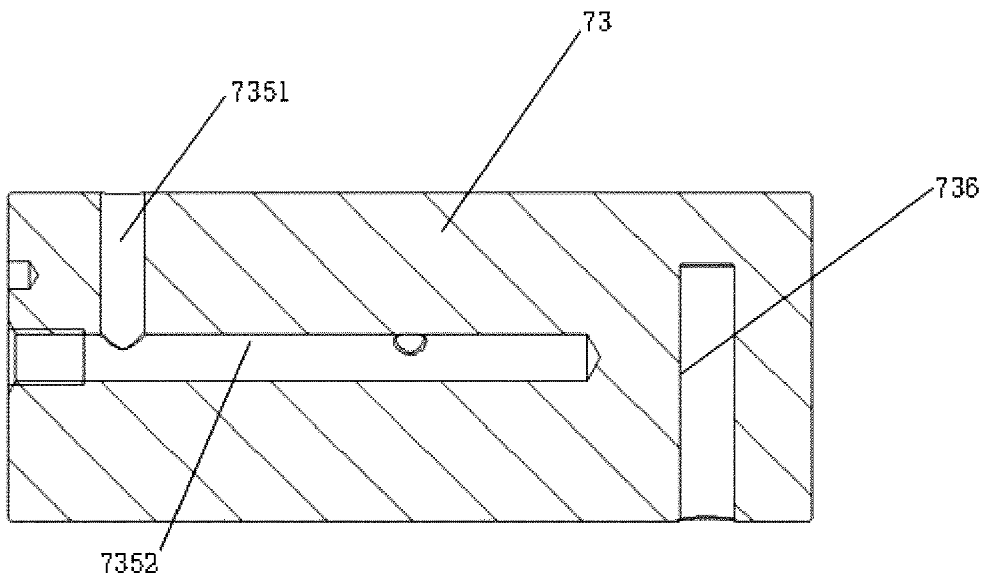

FIG.18

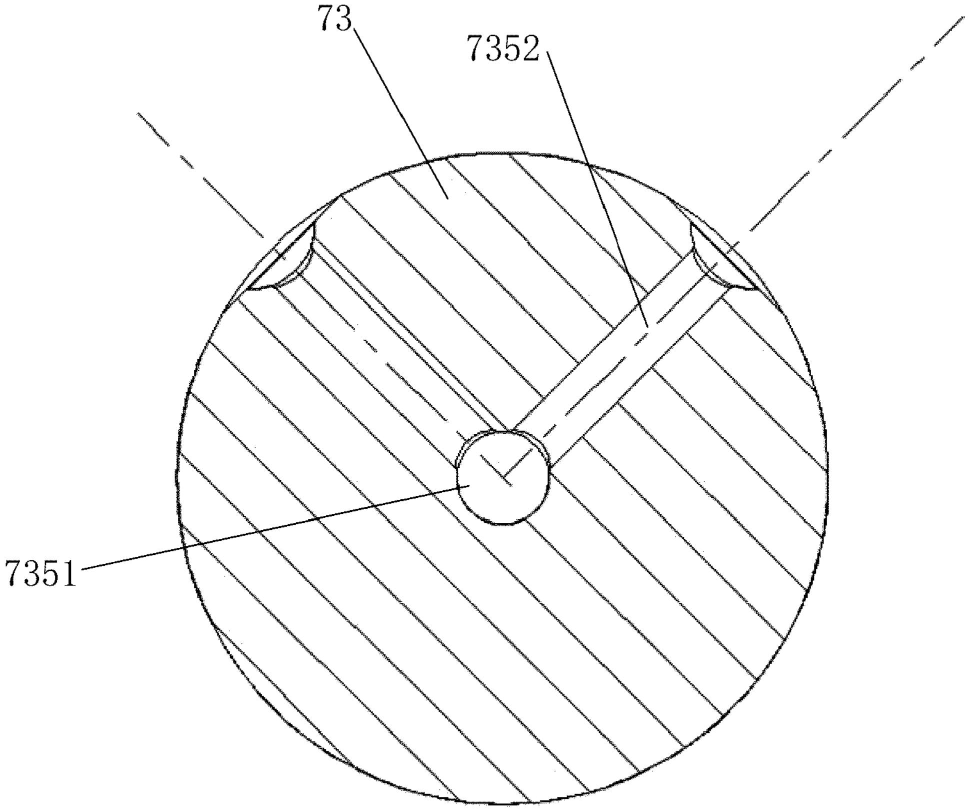

FIG.19

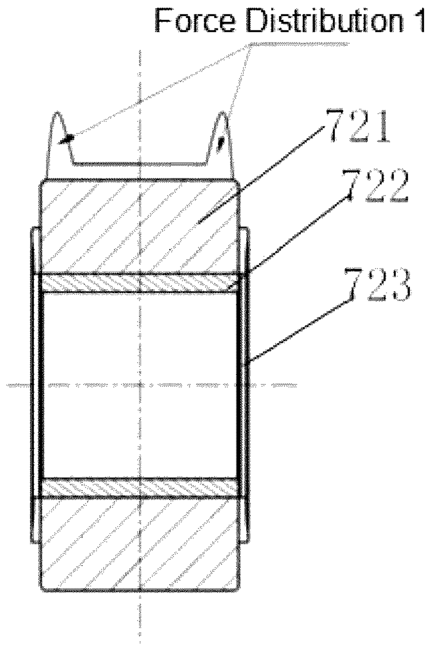

FIG.20a

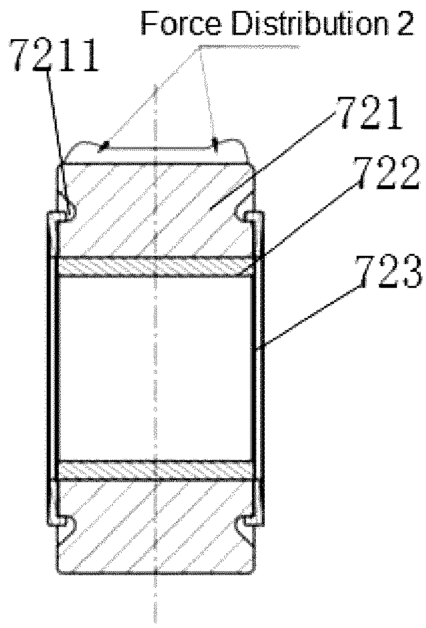

FIG.20b

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EUROPEAN SEARCH REPORT

Application Number
EP 20 20 5958

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Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A	CN 105 888 905 A (CHANGZHOU BOST FUEL INJECTION EQUIPMENT CO LTD) 24 August 2016 (2016-08-24) * paragraph [0032] - paragraph [0034]; figure 3 * * abstract * -----	1-15	INV. F02M53/00 F02M59/10 F02M59/36 F02M59/44 F02M59/46 F02M63/00
A	DE 10 2006 040468 A1 (MAN DIESEL SE [DE]) 6 March 2008 (2008-03-06) * paragraph [0001] - paragraph [0015]; figure 1 * * abstract * -----	1-15	
A	WO 2013/035132 A1 (TOYOTA MOTOR CO LTD [JP]; USUI TAKASHI [JP] ET AL.) 14 March 2013 (2013-03-14) * abstract; figure 3 * -----	1-15	

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F02M

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
The Hague	5 March 2021	Hermens, Sjoerd

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05-03-2021

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